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MECHANICAL REDUCTION GEARS.

BY J. A. DAVIES.

Mechanical reduction gears of the toothed gearing type have been used by engineers for many years. The application of such gearing for the propulsion of ships is of quite recent date. It is true that about the middle of the last century spur gears had been used on board ship. These were increasing gears to permit the use of the early slow-speed reciprocating engines and the higher-speed propellers. Rapid progress in the design of both engines and propellers rendered such gearing unnecessary. Gearing was not again used for the propulsion of ships of large capacity until after the steam turbine had been developed and applied to marine work. The limitations of the direct-connected turbine prohibited its use on any but the fastest ships. In 1910 the Westinghouse Machine Company in the United States built two 3,000 H.P. single-pinion reduction gears for the twin screw U. S. S. *Neptune*. During the same year the Parson's Marine Steam Turbine Company

in Great Britain equipped the single-screw ship *Vespasian* with a two-pinion reduction gear of 1,100 H.P. From that date the progress of reduction gears for marine propulsion has been very rapid. Gears have been built and are building for powers exceeding 22,000 H.P. per gear.

While no fixed rules can be made at the present stage in the progress of reduction-gear design, sufficient has been accomplished to indicate the more important dimensions now in favor among gear designers.

All of the gearing now in use on board ship is of the double-helical type, a right-hand and left-hand helix being used to balance the end thrust which is present in such gearing. The involute tooth contour is universally used for such work. This is largely due to the desirable advantage possessed by involute teeth in that the correct action of the teeth is unimpaired if the centers of the pinion and gear wheel become slightly increased, as might occur due to wear of bearings.

The angle of helix used varies between 23 degrees and 45 degrees. There will be more teeth in contact with the latter angle than with the former, assuming the same circular pitch in each case. On the other hand, the time required, and consequently the cost for cutting teeth of the greater angle, is greater than that for the smaller angle, assuming the same type of cutter in each instance. It would appear that a helix angle of about 30 degrees is the most desirable one from all points of view.

There are two distinct types of mechanical gearing now in use. One is referred to as the fixed or rigid-bearing type, while the other is known as the flexible or floating-frame gear. The latter is an invention of the late Rear Admiral George W. Melville, U. S. N., and Mr. John H. Macalpine. It is built in this country by the Westinghouse Electric & Manufacturing Company. The rigid type of gearing is best known in the marine field from the work of the Parson's Steam Turbine Company of England. The recent unprecedented demand for

geared turbine machinery for ship propulsion has brought into the marine field such other well known reduction gear manufacturers as The De Laval Steam Turbine Company, Trenton, N. J., and The Falk Gear Company, Milwaukee. The General Electric Company of Schenectady have also built a large number of marine reduction gears of the fixed-bearing type, although many of the earlier gears of this company possessed a flexible feature manufactured under the "Alquist" patents.

Experience indicates that extreme accuracy of tooth cutting and the maintenance of accurate alignment are essential to the successful operation of any reduction gear whatever the type employed. In rigid gearing the alignment is determined by accurate machine work in boring the gear housings and fitting the bearings, while in gears of the floating-frame type the pinion and its bearings are mounted in a frame supported on flexible I beams, and automatically maintains its correct position after being accurately aligned in the first instance.

The important feature which must ever be borne in mind is to so design the gearing and its housing that the total tooth pressure shall be uniformly supported across the whole width of the tooth face. Misalignment will prevent this uniform distribution. So also will excessive bending deflection of the pinion, as frequently occurs where the diameter is small and the tooth face long. This is particularly true of pinions supported by two bearings. Occasionally, where the pinion diameter is large, two bearings can be employed, but in most instances a third bearing placed between the separated helices is essential. In every case it is desirable.

As the alignment must be maintained with undiminished accuracy the design of the bearings in a reduction gear must receive proper consideration. Each detail will be separately treated and its functions briefly discussed. Useful formulæ will be given to aid the designer. These formulæ represent present practice so far as has been disclosed by existing designs.

In the formulæ the abbreviations used are as follows:

S.H.P. = Shaft horsepower per pinion.
T.P. = Total tooth pressure on each pinion, pounds.
R.P.M. = Revolutions per minute of gear wheel.
r.p.m. = Revolutions per minute of pinion.
P.C.D. = Pitch-circle diameter of gear wheel, in inches.
p.c.d. = Pitch-circle diameter of pinion, in inches.
L = Total axial length of tooth face; *i.e.*, length of R.H. helix, plus length of L.H. helix.
D.P. = Diametral pitch of hob.

PINION.

If the S.H.P. is to be transmitted by a single pinion and gear wheel, as is the case when one complete expansion turbine per shaft is used, the following formulæ are applicable:

(1) p.c.d. $= C \times \sqrt{\text{S.H.P.} \div \text{r.p.m.}}$, where C = 7.37 for fixed bearing gears and 6.72 for floating-frame gears.

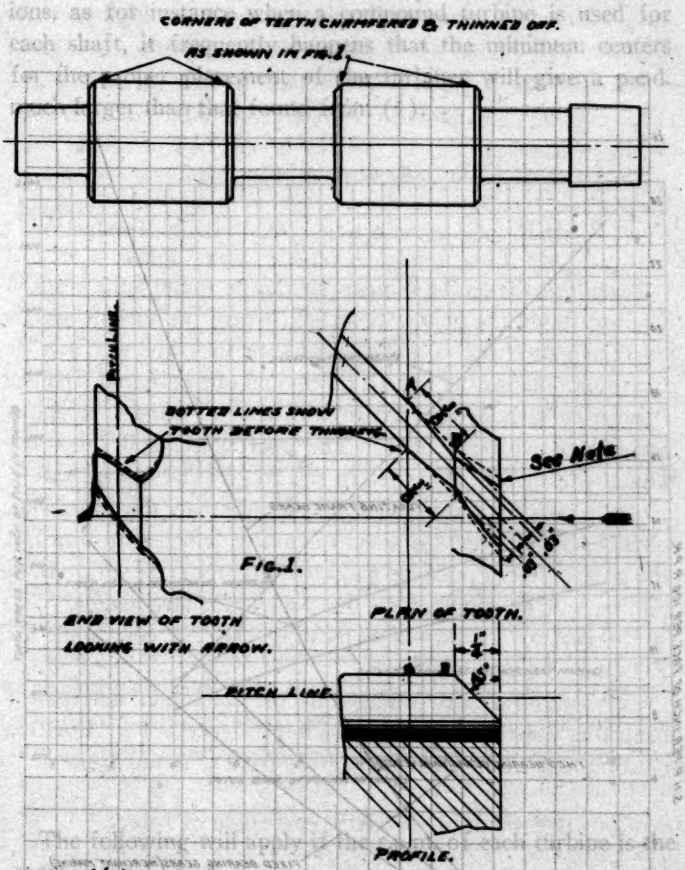
(2) $L = TP \div 175 \sqrt{\text{p.c.d.}}$ for fixed bearing gears, merchant marine.

$L = TP \div 220 \sqrt{\text{p.c.d.}}$ for fixed bearing gears, naval vessels.

L = p.c.d. × 4 for floating-frame gears.

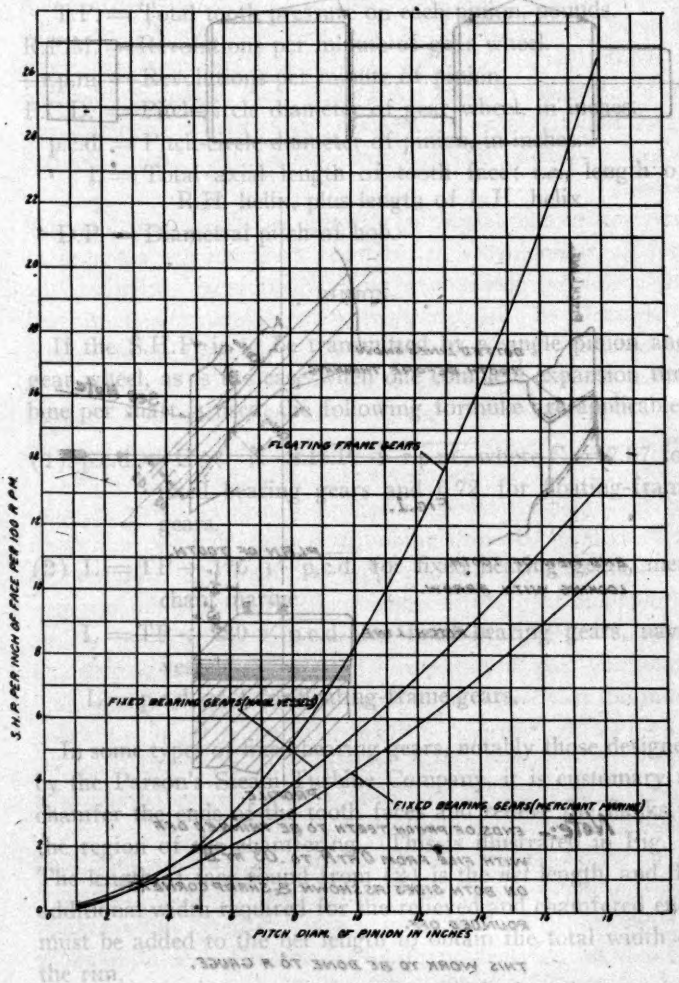
In some types of fixed bearing gears, notably those designed by the Parson's Steam Turbine Company, it is customary to chamfer the ends of the tooth faces and relieve the flanks in the region of the chamfering. This is illustrated in Fig. 1. The length of face found from (2) is the *net* length, and the additional width required for the relieved and chamfered ends must be added to the net length to obtain the total width of the rim.

(3) T.P. = S.H.P. × 126,200 ÷ (p.c.d. × r.p.m.)

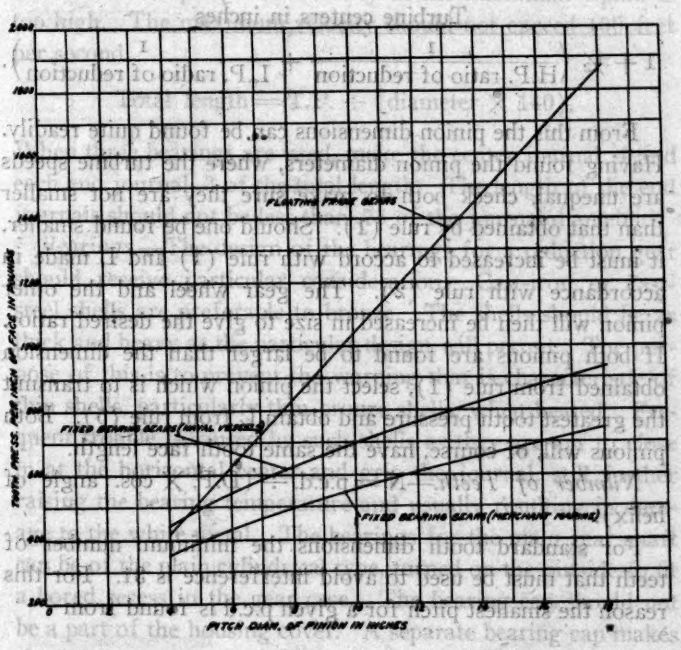


Note:— ENDS OF PINION TEETH TO BE THINNED OFF WITH FILE FROM 0" AT "A" TO .03" AT "B" ON BOTH SIDES AS SHOWN & SHARP CORNERS ROUNDED OFF.

THIS WORK TO BE DONE TO A GAUGE.


FLOATING FRAME GEARS
FIXED BEARING GEARS (NAVAL VESSELS)
FIXED BEARING GEARS (MERCHANT MARINE)
S.H.P. PER INCH OF FACE PER 100 R.P.M.
PITCH DIAM. OF PINION IN INCHES
0
2
4
6
8
10
12
14
16
18
20
22
24
26

In a single reduction gear of low ratio driven by two pinions, as for instance when a compound turbine is used for each shaft, it frequently happens that the minimum centers for the proper placement of the turbines will give a p.c.d. much larger than that found from (1).


FLOATING FRAME GEARS
FIXED BEARING GEARS (METAL PINIONS)
FIXED BEARING GEARS (RAWHIDE PINIONS)
TOOTH PRESS. PER INCH OF FACE IN POUNDS
PITCH DIAM. OF PINION IN INCHES

The following will apply if the r.p.m. of each turbine is the same:

(4) p.c.d. = dimension of pinion centers in inches ÷ (ratio + 1).

(5) L = T.P. ÷ (p.c.d. × 104) for floating-frame gears. For fixed bearing gears the formulæ given in (2) apply. It is desirable to keep the minimum dimension of L = p.c.d. × 2.

If the r.p.m. of the two turbines are different, which is the case with many compound turbines where the high-pressure element has higher r.p.m. than the low-pressure, the gear-wheel diameter is found from the following:

$$\text{P.C.D.} = \frac{\text{Turbine centers in inches}}{1 + \frac{1}{2}\left(\frac{1}{\text{H.P. ratio of reduction}} + \frac{1}{\text{L.P. radio of reduction}}\right)}.$$

From this the pinion dimensions can be found quite readily. Having found the pinion diameters, where the turbine speeds are unequal, check both to make sure they are not smaller than that obtained by rule (1). Should one be found smaller, it must be increased to accord with rule (1) and L made in accordance with rule (2). The gear wheel and the other pinion will then be increased in size to give the desired ratios. If both pinions are found to be larger than the dimension obtained from rule (1), select the pinion which is to transmit the greatest tooth pressure and obtain L from rule (5). Both pinions will, of course, have the same tooth face length.

Number of Teeth.—N = p.c.d. ÷ (D.P. × cos. angle of helix).

For standard tooth dimensions the minimum number of teeth that must be used to avoid interference is 31. For this reason the smallest pitch for a given p.c.d is found from

$$\text{D.P.} = \text{p.c.d.} \div (31 \times \text{cos. angle of helix}).$$

If the D.P. thus found is not a standard size, increase the number of teeth and use the next smaller D.P. The p.c.d. may also need slight modification to permit the use of standard cutters. Having settled the p.c.d. the other tooth dimensions can be found as follows:

Diameter at top of teeth = p.c.d + (2 ÷ D.P.).

Diameter at root of teeth = p.c.d. − (2.314 ÷ D.P.).

Pinion Journals.—Maximum diameter = p.c.d. — (2.470 ÷ D.P.).

The diameter thus obtained brings the surface of the journal as close to the root of the teeth as it is practically desirable to go. Nothing is to be gained by reducing the diameter below this unless the peripheral velocity at the maximum r.p.m. is too high. The maximum velocity should not exceed 100 feet per second.

$$\text{Total length} = \text{T.P.} \div (\text{diameter} \times 140).$$

When three bearings are used, make the center journal .4 and each end journal .3 of the total length. The length of the end journals should not be less than .75 of the journal diameter.

Bearings.—The design of the bearings for a reduction gear should receive particular consideration. Cast-iron or cast-steel shells are preferable to bronze. The shells should be as thick and heavy as the particular design will permit. The purpose of this is to prevent the warping that is characteristic of thin shells, particularly thin bronze shells, when heated. Frequent trouble is caused by such shells as they are apt to close in at the horizontal center and grip the journal, still further raising the bearing temperature and usually resulting in damage to the white metal. The bearings for the main gear shaft can be of the plain cylindrical type, turned on the outside to fit a bored recess in the gear case. The bearing cap should not be a part of the housing cover. A separate bearing cap makes the bearing more accessible and allows of adjustment when fitting the shell to its recess. The shell must be tightly held in the housing. The pinion bearings should be provided with means of adjustment so that the correct running centers of the gears can be set without difficulty. In some of the early gears of the fixed-bearing type, the pinion bearing shells were of the plain cylindrical type and entire reliance had to be placed on the accuracy of the machine work of the gear housing for the satisfactory operation of the teeth. While it is possible to bore a gear housing so that two sets of bearings

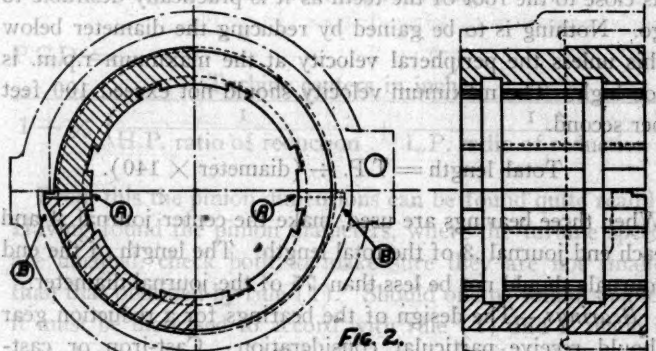


FIG. 2.

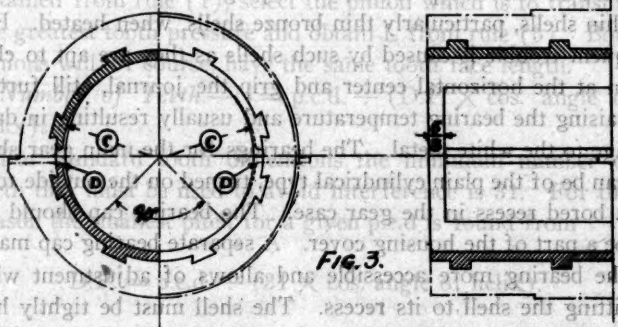


FIG. 3.

A. A.—Oil inlet slots on each side.
B. B.—Lugs on each side to prevent turning.
C. C.—White metal eased away by counterboring at each side, this to extend to within ½ inch of each end and to leave 90° of surface at top and bottom.
D. D.—Oil passages to be well rounded.

are parallel and in the same plane, the most practical method is to provide some adjustment to one set of the bearings and permit the gear centers to be accurately placed when assembling. Furthermore this will be of distinct advantage when spare bearings have to be fitted on board ship.

The lining for both pinion and gear bearings should be of good quality of tin-base white metal. Lead-base mixtures have been frequently used for the lower speed gear bearings, but such metals are undesirable owing to the extreme care necessary when melting. The bearing shells should be provided with adequate anchor grooves for holding the white metal, which should be well peened after it is poured. While it may be possible to core the anchor grooves in the bearing shells it is generally found desirable to rough bore the shell and plane the longitudinal anchor grooves. The latter grooves are frequently dovetailed as indicated in Fig. 2, while the circumferential grooves can be cored with parallel sides. The figure shows a suitable construction for a cylindrical bearing. The side lugs on the lower half project under the bearing cap to prevent the shell from turning.

The surface of the white-metal lining must be machined and fitted so that the lubricating oil can have easy access to the whole surface. It should be borne in mind that the oil has to cool as well as lubricate the journal. The first duty demands large, well rounded oil passages and a copious supply of oil. The second is entirely dependent on the formation and maintenance of an oil film separating the steel journal and white-metal lining. This is all important in the bearings of modern geared turbines, the journals of which often operate at peripheral velocities as high as 100 feet per second. In such bearings the destruction of the oil film, due to failure of the oil or a faulty bearing surface, would result in rapid heating and wiping of the white metal. In a reduction gear this would immediately alter the alignment of the pinion and gear wheel. For this reason the designer should determine

the clearances and the shape of the finished white metal surface and not leave such important details to be settled in the machine shop. A clearance of .002 inch per inch of diameter should be given to pinion and gear bearings. The clearance on all of the pinion bearings must be equal, for the reason that the pinion changes from the lower to the upper half of the bearing or vice versa, when going from ahead to astern. Obviously if the clearances were not equal the pinion would change its alignment with the gear wheel in such circumstances.

Suitable proportions for the white-metal lining will be found in Fig. 3.

Pinion Couplings.—The coupling fitted between the turbine and gear should be designed to permit an axial movement of the pinion so that the load can be equally divided between the right-hand and left-hand helices. Two types are in common use. One in which round pins on the turbine coupling head drive in bronze bushed holes in the pinion coupling flange. The other has claws or projections on each half of the coupling. Either type requires copious lubrication for satisfactory service, and the workmanship necessary is of a high order. All of the pins or claws must bear equally when transmitting power either ahead or astern. The pin type is the cheapest to produce. Greater accuracy of machine work is also possible with this type. Figs. 4 and 5 show typical examples of the two types.

Flexible Driving Shafts.—While the foregoing indicates two couplings possessing end freedom, they are not, strictly speaking, "flexible" couplings. This term is a misnomer when applied to couplings. Many so-called "flexible" couplings are on the market. The principal claim for each is that misalignment between the pieces coupled will be cared for by the feature of flexibility peculiar to the coupling used. These couplings are usually undesirable in high-powered, high-speed machinery. In any case the alignment of such machinery should be made and checked with the closest accuracy. No excuse can be tolerated for inferior workmanship in this feature. The

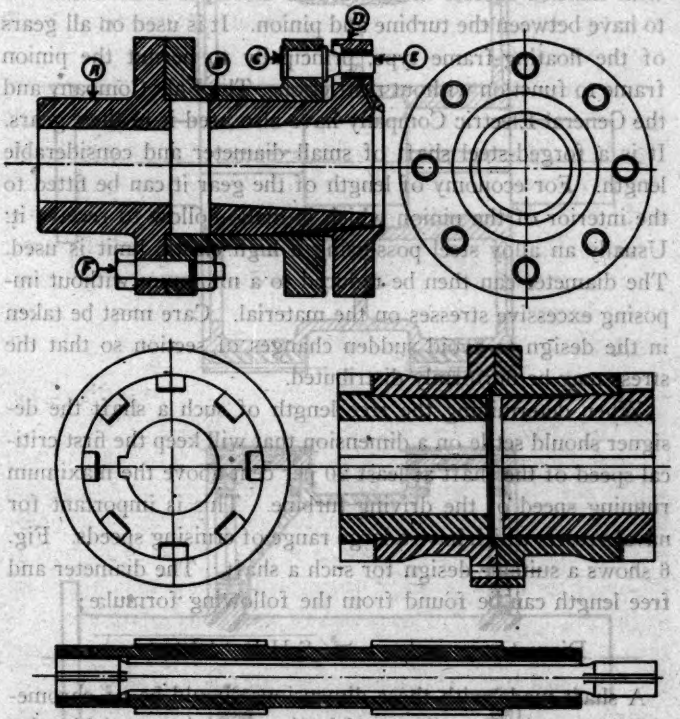


FIG. 4.—PIN COUPLING.

FIG. 5.—JAW COUPLING.

FIG. 6.—HOLLOW PINION WITH INTERNAL FLEXIBLE DRIVING SHAFT.

Nomenclature of Fig. 4.

A.—Coupling on Pinion Shaft.
B.—Sliding Fit.
C.—Riveted Over.
D.—Coupling on Turbine Shaft.
E.—Coupling Pins in Bronze Bushed Holes.
F.—Fitted Bolts.

closer the alignment the more perfect will be the operation. It is, however, possible to augment the pin or claw-type coupling with a flexible shaft, and this feature, which De Laval used with marked success in his turbine drives, is a desirable link to have between the turbine and pinion. It is used on all gears of the floating-frame type, principally to permit the pinion frame to function without restriction. The Falk Company and the General Electric Company have also used it in their gears. It is a forged-steel shaft of small diameter and considerable length. For economy of length of the gear it can be fitted to the interior of the pinion which is made hollow to receive it. Usually an alloy steel possessing a high elastic limit is used. The diameter can then be reduced to a minimum without imposing excessive stresses on the material. Care must be taken in the design to avoid sudden changes of section so that the stress may be uniformly distributed.

When determining the free length of such a shaft the designer should settle on a dimension that will keep the first critical speed of the shaft at least 20 per cent above the maximum running speed of the driving turbine. This is important for naval vessels which have a large range of cruising speeds. Fig. 6 shows a suitable design for such a shaft. The diameter and free length can be found from the following formulæ:

$$\text{Diameter in inches} = \sqrt[3]{\text{S.H.P.} \times 18 \div \text{r.p.m.}}$$

A shaft made with these dimensions should be of chrome-nickel or equivalent alloy steel having a minimum yield point of 80,000 pounds per square inch and an elongation of 20 per cent minimum in 2 inches. Free length of shaft, in inches,=

$$\sqrt{2210 \times 10^3 \times \text{diameter in inches} \div (\text{r.p.m.} \times 1.20)}.$$

GEAR WHEELS.

The large wheels of a reduction gear are usually built up of three pieces, a shaft of forged steel, a center of cast steel or cast iron and rims of forged or rolled steel. Occasionally

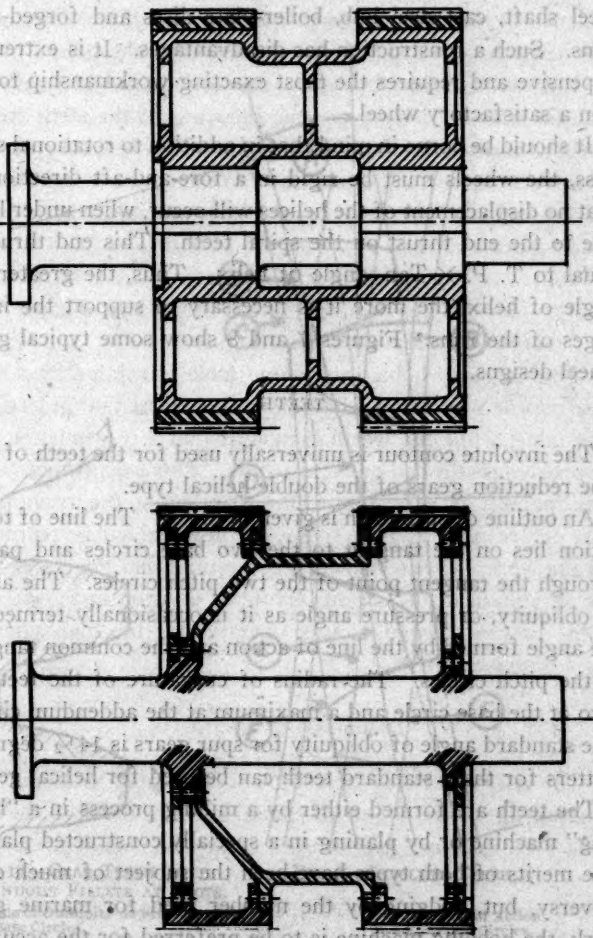


FIG. 7.—GEAR WHEEL, ONE PIECE.

FIG. 8.—GEAR WHEEL, BUILT UP.

where low weights are of importance, as for instance in destroyer work, the wheels have been built up from a forged-steel shaft, cast-steel hub, boiler-plate discs and forged-steel rims. Such a construction has disadvantages. It is extremely expensive and requires the most exacting workmanship to obtain a satisfactory wheel.

It should be borne in mind that in addition to rotational stiffness, the wheels must be rigid in a fore-and-aft direction so that no displacement of the helices will occur, when under load, due to the end thrust on the spiral teeth. This end thrust is equal to T. P. $\times$ Tan. angle of helix. Thus, the greater the angle of helix, the more it is necessary to support the inner edges of the rims. Figures 7 and 8 show some typical gear-wheel designs.

TEETH.

The involute contour is universally used for the teeth of marine reduction gears of the double helical type.

An outline of such teeth is given in Fig. 9. The line of tooth action lies on the tangent to the two base circles and passes through the tangent point of the two pitch circles. The angle of obliquity, or pressure angle as it is occasionally termed, is the angle formed by the line of action and the common tangent to the pitch circles. The radius of curvature of the teeth is zero at the base circle and a maximum at the addendum circle. The standard angle of obliquity for spur gears is 14½ degrees. Cutters for these standard teeth can be used for helical gears.

The teeth are formed either by a milling process in a "hobbing" machine or by planing in a specially constructed planer. The merits of both types have been the subject of much controversy, but, judging by the number used for marine gear work, the hobbing machine is to be preferred for the accurate work that is necessary in high-speed gearing. The cutting is done by a milling tool termed a "hob." It is a worm, in the thread of which approximately longitudinal gashes are made to form cutting edges. The wheel to be cut is mounted on a

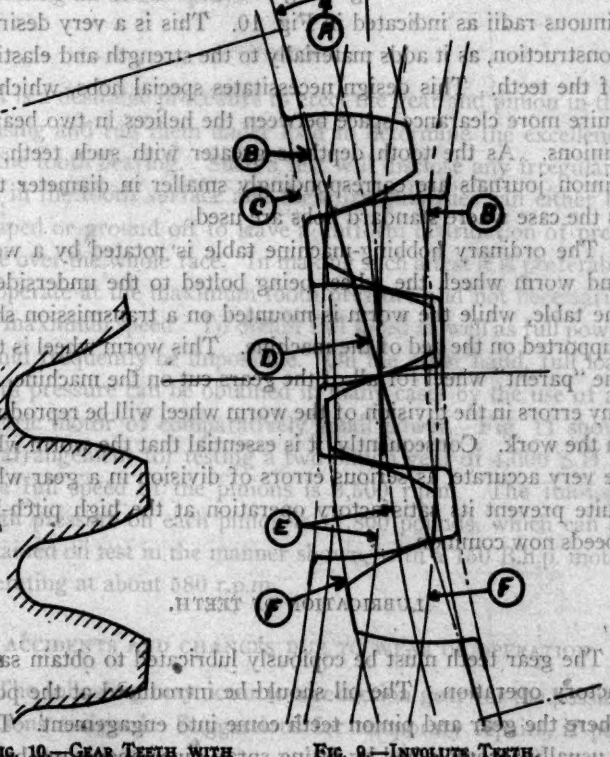


FIG. 9.—INVOLUTE TEETH.

FIG. 10.—GEAR TEETH WITH CONTINUOUS FILLETS AT ROOTS.

A.—Angle of Obliquity or Pressure Angle.
B, B.—Base Circles.
C.—Root or Dedendum Circle.
D.—Line of Action.
E.—Pitch Circles.
F, F.—Addendum Circles.

revolving table, which is usually horizontal, and the hob is fed vertically across the face of the gear rim. The helix angle is formed by establishing the correct rate of travel of the table carrying the work and the head supporting the hob.

It is essential that the root of the teeth be provided with substantial fillets. In some gearing the roots of the teeth are continuous radii as indicated in Fig. 10. This is a very desirable construction, as it adds materially to the strength and elasticity of the teeth. This design necessitates special hobs, which require more clearance space between the helices in two bearing pinions. As the tooth depth is greater with such teeth, the pinion journals are correspondingly smaller in diameter than is the case where standard hobs are used.

The ordinary hobbing-machine table is rotated by a worm and worm wheel, the wheel being bolted to the underside of the table, while the worm is mounted on a transmission shaft supported on the bed of the machine. This worm wheel is thus the "parent" wheel for all of the gears cut on the machine, and any errors in the division of the worm wheel will be reproduced in the work. Consequently, it is essential that the worm wheel be very accurate, as serious errors of division in a gear wheel quite prevent its satisfactory operation at the high pitch-line speeds now common.

LUBRICATION OF TEETH.

The gear teeth must be copiously lubricated to obtain satisfactory operation. The oil should be introduced at the point where the gear and pinion teeth come into engagement. This is usually accomplished by fitting spray pipes above and below the pinion to lubricate both the ahead and astern faces. Successful practice indicates that a few large holes are preferable to a large number of small spray holes, as the latter are quite easily choked by small particles of lint which occasionally get past the strainers. In arranging these sprays care should be taken to firmly support any pipes that may be used, to prevent

fracture due to vibration. On some gears the sprays are individually arranged so that they can be removed and cleaned, if necessary, while the gears are in operation. As such a design involves the use of valves in the oiling system, particular care must be exercised in the arrangement to prevent any possibility of closing all of the sprays.

TESTING.

It is a desirable procedure to erect the gear and pinion in the housing and run them under load to determine the excellence of the tooth bearing. Such a test will disclose any irregularities in the tooth surface and these irregularities can either be scraped or ground off to leave a uniform distribution of pressure over the whole face. In making such a test it is preferable to operate at the maximum tooth pressure and not necessarily the maximum speed. To obtain full speed as well as full power would frequently be impossible. On the other hand, full load tooth pressure can be obtained in many cases by the use of an electric motor of comparatively small power. Fig. 11 shows an arrangement for testing a two-pinion gear of 4,000 S.H.P. The full speed of the pinions is 3,500 r.p.m. The full-load tooth pressure on each pinion is 11,800 pounds, which can be obtained on test in the manner shown, with a 150 B.h.p. motor operating at about 580 r.p.m.

ACCIDENTS AND CHANGES DUE TO WEAR OR OPERATION.

The following may occur to a reduction gear for the possible reasons assigned. Suggestions for temporary repairs, if possible, are given:

Rough Handling.—It is perhaps needless to state that the utmost care must be used in handling the gears to prevent damage to the teeth. The teeth are small in pitch and because of this they are more liable to injury if roughly handled. Should the tops of some of the teeth receive a blow of sufficient

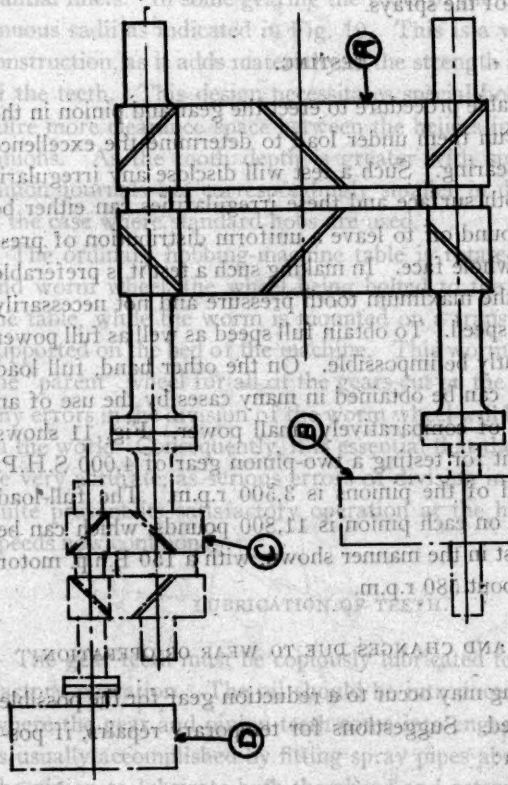


FIG. 11.

A.—Gear to be Tested.
B.—Prony Brake.
C.—Small Reduction Gear for Test Purposes.
D.—Electric Motor.

force to dent the metal, the flanks of the teeth so damaged, in the region of the damage, will be spread, increasing the thickness of the teeth at this point. It will therefore be necessary to scrape both sides of the teeth for a short distance in this vicinity, otherwise a decidedly high spot will be revealed after the gear has been in operation.

Damaged Gear Teeth.—This may be caused by a foreign body, such as a nut or a wrench, being left in the gears or by a defective tooth breaking out. For such an accident it is hard to prescribe beforehand. One thing is sure, and that is both the pinion and gear wheel teeth will be spoiled at the damaged section. If the damage has occurred at the ends of the faces, the damaged part of the tooth face on the pinion can be turned off to just below the root. This will enable the undamaged portion of the teeth to work together without interference from the damaged parts of the gear-wheel teeth and avoids the necessity for handling the gear wheel. The pinion can then be replaced and put into operation until such time as a new pinion and wheel are obtained.

Pitting.—During the first few months of service there has often been observed on the flanks of the gear teeth of large gears an action known as "pitting." Small flakes of metal break out of the otherwise perfect surface of the teeth, leaving hollow depressions. These depressions, or pit marks, vary in size from the smallest possible mark to as large as 3/16 inch diameter and fifteen to thirty thousandths of an inch deep. They may appear in any part of the tooth flank from the tip to the root, but as a general average it can be said to be most common in the region of the pitch line where only rolling action occurs. It is difficult to say whether this is merely a coincidence, as the cause does not appear to be due to this rolling action. A lack of homogenity in the material at the tooth surface or crushing of minute high spots causing fatigue and failure of small sections of the metal at such high spots are possible reasons for this pitting.

Experience shows, however, that any pitting which may take place usually occurs during the first few months of service and then entirely ceases. No trouble of any description is experienced during this period and the operation of the gear is not in any way delayed or affected.

Scoring of Tooth Surfaces.—Should the lubricating-oil supply to the gear teeth sprays fail, the teeth will score and the polished flanks tear into a rough damaged surface. The absolute need for an uninterrupted supply of clean oil for the satisfactory operation of the teeth cannot be emphasized either too strongly or too often. If, after all precautions have been taken, the oil supply should fail and the teeth become scored, the gears must be thoroughly overhauled at the first opportunity. The tooth surfaces should be smoothed down with a carborundum stone and possibly scraped. If the scoring is severe the roughened surfaces may need lightly filing to remove excessive burrs and ridges. The use of a file on a tooth shape must be done with care and judgment and only a well-trained mechanic should be entrusted with the work.

The oiling system will need a thorough cleaning as flakes of steel from the damaged teeth will be found mixed with the oil.

If, after oil failure, the teeth are found to be scored and it is inexpedient to make an immediate overhaul, the oil can be restored and the gears operated without fear of breakage. This fact must not be used to delay proper repairs at the earliest moment.

Corrosion.—The gear teeth and journals may show signs of rust with resulting corrosion. This is invariably due to an acid lubricant, or because of a salt-water leak into the lubricating system, and can only be remedied by changing the oil. No harm will be experienced providing the cause is eliminated without undue delay. The corroded journals should be stoned with an oil stone and the teeth cleaned with a fine carborundum stone.

Wear of Teeth.—Under normal conditions the gear and pinion teeth should not wear a measurable amount in a considerable period of time. It is not usual to keep records of the wear of gear teeth, as experience justifies the statement made in the foregoing sentence. It is natural to expect the greatest wear, if any, to occur on the pinion teeth, but any periodic measurements which might be taken should show very small variations. No excessive wear can take place without scoring, as an oil film prevents metallic contact between the surfaces of the teeth.

VENTILATING AND HEATING FROM THE MARINE POINT OF VIEW.

By CHAS. F. GROSS, ASSOCIATE.

The ventilation of merchant ships is as large a problem as the ventilation of an office building. The system is less complicated than is required on board a battleship. The purpose of ventilation is to remove air which has become hot, moist, or contaminated with undesirable elements and the replacement with pure air. The replacement air must fulfill certain requirements depending on the place to be ventilated. The air should be delivered to a compartment at a low velocity such that it does not create a great draft. For good circulation the air should have a velocity of about eight feet per second. Sometimes this air is made to circulate through the ship by reason of the ship's motion through the water. A vessel steaming nine knots per hour creates an air current of fifteen feet per second, while a fast ocean liner at twenty-one knots produces a current of circulating air of thirty-five feet per second. When the ship is steaming with the wind it does not get the benefit of this artificial draft.

The required temperature of the new air depends upon the compartment which is to be ventilated; that is, for use in a cabin, stateroom, storeroom, cargo space or workshop. The amount of moisture in the air is another important quantity and this depends upon the temperature. Dry air at 80 degrees F. has about four times the capacity for moisture that dry air at 40 degrees F. has. The air must contain the correct amount of moisture so that it will not have a disagreeable cooling effect. Conditions for obtaining more perfect ventilation are better at sea than ashore. Sea air contains about 3 cubic feet of carbonic acid gas to 10,000 cubic feet of air. The per cent content of this gas is taken as the best test for the purity of

the air, as its presence also indicates the presence of other undesirable elements. The cubic feet of air necessary for the average person per hour varies depending on the purity of the air after exhalation. If the purity is to be low (high CO_2 percentage) the air necessary will be about 500 cubic feet per hour, whereas, if the purity is to be high, the air must be replaced at the rate of about 2,500 cubic feet per hour.

Ventilation on board ship is accomplished in two ways, by natural draft and by mechanical means. The natural system is briefly defined as the circulation of air by either the tendency of hot air to ascend and give place to heavier cold air or by directing air currents, which are caused by the vessel's motion, through desired spaces. The mechanical system depends upon a circulation by fan or blower. In fine weather, ventilation by natural means may be all that is desired, but in stormy weather, where large openings must be closed the circulation may become impaired; this may not be serious except in the tropics. For staterooms the ventilators should be such that the air be replaced at the rate of five to six times per hour. This is mostly accomplished by having ornamental openings at the top and bottom of the partitions whereby the air can circulate freely into the passage ways. These passage ways lead directly to the machinery spaces and afford an excellent circulation of air. Cabins on the outboard side may be thoroughly ventilated by side lights. On some ships, side lights are installed that will permit light and air to enter at all times, floats and valves are installed which operate when submerged and prevent water from entering the ship. When cabins have open deck space above them, they have individual mushroom or ejector ventilators which are small but very efficient. Storerooms are usually located below decks, and the air should be renewed between five and six times per hour. They are ventilated by having spaces between two frames enclosed by a piece of sheet iron, thereby forming a duct which is led to a weather deck, terminating in a goose neck. Coal bunkers are ventilated by leading a large duct from the bunker to the

space around the inner and outer stacks, the air being supplied to the bunkers by goose necks. The shaft tunnel is ventilated very efficiently by placing a cowl at the most extreme end and turning it so that it acts as an uptake. Air is supplied to the tunnel from the downcasts in the engine room. This cowl must be watertight and its construction heavier than ordinary ones, as recent developments have shown that the tunnel may be filled with water. Double bottoms and peak tanks are ventilated by goose necks attached to the air-vent pipes. Only when the tanks remain empty for any length of time does the air become vitiated.

Machinery spaces need large volumes of fresh air, the change being about 25 or 30 times per hour. The boiler room has usually two or four cowls, the upper ends being close to the stack while the lower end of the cowl leads to within eight feet of the firing platform. The cowls have a capacity considerable in excess of what is needed for combustion; the surplus cools the stokehold and forces the heated air out through the fidley. These cowls are placed on the fidley grating and are provided with gear and pinion so that they can be turned from the firing platform. At times when there is little wind on deck, especially when at anchor, little air goes down the cowl (just enough for combustion), and the fireroom may become very hot. Frequently blowers are installed to assist the cowls, and no attempt is made to build up the pressure in the fireroom as the upper portion is open to the outside atmosphere through the fidley which allows the heated air to escape. The design of the deck intakes with their cowls and intakes has been so greatly influenced by the requirements of protection from waves and spray in bad weather that less attention has been given to the proper admission of air without loss or restriction. Frequently the area of the opening is insufficient or not effective. The difficulty of getting air into the shaft leading to the boiler room becomes greater as the speed of the ship increases. Many types of deck intakes, cowls and weather flaps have been tested and it has been found that inlets

should have the correct clearance between the vanes, correct angle of entry and the avoidance of sharp corners. It is generally thought that the air delivered to the boiler rooms is uniformly distributed, but the lifts and compartments reaching down into the boiler room add considerable to the difficulty of getting air thoroughly circulated. For the engine room there are usually two cowls, one so placed as to discharge close to the throttle, both acting as down casts. The heated air escapes out through the machinery hatch and shaft tunnel. Here the cowls should have a capacity to change the air from 15 to 20 times per hour.

Artificial or mechanical ventilation is effected by either of two systems; the plenum and the exhaust. Briefly the first consists in forcing a quantity of pure air into a certain compartment, thereby replacing a certain quantity of vitiated air. In the exhaust method fans draw the foul air from the compartments and exhaust it to cowls. Toilets, kitchens and rooms where smoke, dust or gases accumulate should be ventilated by the exhaust method. As a whole the plenum system is preferable as it is more positive. The exhaust system is considered more healthful, as foul air is taken from the compartment and discharged outside instead of being driven through open spaces of the vessel, as it passes freely from each compartment to the outside air. In the layout of any system the amount of air to be delivered to each compartment is first determined. It has been found that the following should be observed in designing on the plenum ventilating system: Starting with a fan discharge of about 2,000 cubic feet per minute and a pressure of about 5 pounds per square foot, the mains should be round, if possible, and maintain an original diameter as long as the velocity does not drop below 1,400 feet per minute; then the diameter should be reduced at the rate of one and one-half inches per foot until the velocity is again 2,000 feet per minute. The final diameter of the main should not be reduced to less than twice the diameter of the last branch. All branches should be taken off of the

main duct at an angle of 30 degrees, but near the end of the main the angle should be increased to 90 degrees. For bends, the center line of the pipe should not have a smaller radius than one and one-half times the diameter of the pipe. The effect of the air going around a 90-degree elbow is such that it adds three feet to the effective length of the pipe, for two elbows the length is increased by seven feet and for three by ten feet. The outlets from branch pipes should be enlarged at a rate of one and one-half inches per foot to reduce the velocity at the exit to 1,000 feet per minute. This enlargement causes an increase discharge from the line as much as 25 per cent in long branches.

The ventilators most commonly used aboard ship are of the cowl type made of galvanized iron. The fixed or coaming portion is made of one-quarter to three-eighths-inch steel plating, but if the ventilator is subjected to external pressure it is made thicker and watertight. The classification societies have issued brief instructions for ventilators and these have become standard practice. They require all holds, between decks and coal bunkers, to have at least two ventilators, one a downtake and the other an uptake, placed as far from each other as possible. This insures that the air will circulate freely before leaving the compartment. If the hatches or filling shoots of coal bunkers are fitted with gratings, ventilators may be dispensed with unless the bunker is near the boiler, in which case extra means of ventilation is necessary. A ventilator of the mushroom, swan neck or other approved form must be arranged and led into tanks where gas is likely to gather. Ventilators must be fitted with a fixed coaming to the deck and a removable cowl and should be carried to above a weather deck. Provision must be made for closing the ventilator when the cowl is removed, as in bad weather. The height of the coaming above the weather deck is from two to three feet, according to its diameter. Ventilator tubes should be air-tight below the upper deck. Instead of cowls or ventilators, swan necks, mushrooms or other approved design

may be installed, provided they are strongly fixed to the deck and the opening shows at least two feet above the deck. Mechanical means for supplying fresh air to the compartments is sometimes used in place of downcast ventilators. Ducts passing through crew's quarters should be gastight. Ventilating spaces between decks may be done by the same cowl, the tube leading from the space between decks is made smaller than the coaming to which the cowl is fastened. The between-deck space is then ventilated by the annular space between the tube and the ventilator casing.

If it is desired to install two ventilators to act as downcasts in the engine room to supply 2,000 cubic feet per minute, the question usually is to determine the diameter. The outlet may be 30 feet below the throat of the cowl and the ship have a speed of 12 knots per hour or 20.3 feet per second. This is equivalent to a head of air of

$$\frac{V^2}{2g} = h \text{ or } \frac{20.3^2}{2 \times 32.2} = 6.39 \text{ feet.}$$

By tests it has been found that the velocity of the air going down the ventilator does not increase inversely proportional to the area of the mouth and the throat, but is less at the throat, about 0.6 of the head of the wind at the mouth. The head at the throat will be $6.39 \times 0.6 = 3.83$ feet. The corresponding velocity is

$$\sqrt{2 \times g \times h} = \sqrt{2 \times 32.2 \times 3.83} = 15.7 \text{ feet per second.}$$

$$\frac{2{,}000 \text{ cu. ft.}}{2} = 1{,}000 \text{ cubic feet per cowl.}$$

First a diameter of 15 inches will be assumed. If the tube is straight the head lost by friction will be

$$H_f = 0.00032 \times \frac{L}{D} \times V^2,$$

where L and D are the length and diameter of the tube, re-

spectively, and V is the velocity of the air at the throat in feet per second. In this case

$$H_f = 0.00032 \times \frac{30}{1.25} \times \overline{15.7}^2 = 1.90 \text{ feet friction loss.}$$

Therefore the head at exit or down at the bottom of the ventilator will be 3.83 — 1.90 = 1.93 feet; the velocity corresponding will be $\sqrt{2 \times 32.2 \times 1.93} = 11.1$ feet per second. The volume discharged will be

$$\frac{\pi \times D^2 \times \text{Vel.}}{4} \text{ or } \frac{176.7}{144} \times 11.1 = 13.6 \text{ cubic feet per second.}$$

This is equivalent to 816 cubic feet per minute, showing our assumption of a 10-inch diameter was too small. By using a 16-inch ventilator the delivery would be 996 cubic feet per minute per cowl.

HEATING.

Heating and ventilating are very closely related and one cannot touch on ventilating without saying a little about heating. There are several methods available for heating on board ship, either by steam, water, electricity or air. That mostly used in the marine service is steam, water seldom, and electricity rarely. Steam has the advantage that it may be taken from the auxiliary steam line or, for large work, an independent line is run; in either case it is taken through a reducing valve dropping the steam to a 15-pound pressure. As a safeguard there is installed a safety valve after the reducing valve. It is not good practice to have the heating of cabins and living quarters dependent on each other for their steam supply; sometimes several lines are run to various parts of the ship when living quarters are separated, so that either may be cut in or out independent of the other. Radiator pipes may be installed in two ways; sometimes one pipe alone leads to a radiator, the condensed steam flowing back the same pipe or drawn off by cocks, or, as more common, two pipes are installed. In the two-pipe arrangement one pipe leads steam to the radiator while the second pipe conducts the condensed

steam to steam traps, from whence it is discharged to the main feed tanks in the engine room. Provision is made so that all pipes may be drawn free of water in case of repairs or when the system is laid up. Where compartments are far from the engine room, long leads of steam piping are required. In this case electric heaters may be installed. This is expensive and is only justified in extreme cases. Calculations are sometimes made to determine the square feet of heating surface necessary for a certain room, account being taken of the temperature to which the various sides are subjected and of what material the sides are composed. From tests it has been determined that with a standard make radiator when operated with steam at 212 degrees F. and the surrounding air at 70 degrees the heat transmission is at the rate of 1.57 B.t.u. for one degree temperature difference. The steam condensed per square foot per hour weighs 0.23 pound. Knowing the heat loss through the various sides of the room, the square feet of surface may be determined by the expression:

$$\frac{\text{B.t.u. radiated and conducted away per hour}}{1.57 \times (\text{Diff. in temp. of steam in radiator and outside air})}$$

According to good practice of leading shipbuilding companies, allowance of square feet of radiator surface is proportioned to the cubic contents of the room. The following are good proportions:

Captain's room, 1 square foot per 45 cubic feet.
Captain's bath, 1 square foot per 30 cubic feet.
Pantry, 1 square foot per 85 cubic feet.
Crew's mess, 1 square foot per 35 cubic feet.
Chief engineer, 1 square foot per 43 cubic feet.
Spare room, 1 square foot per 80 cubic feet.
Steward and mess boy, 1 square foot per 66 cubic feet.
Officer's wash-room, 1 square foot per 27 cubic feet.
First and second mate, 1 square foot per 33 cubic feet.
Chart house, 1 square foot per 44 cubic feet.
Pilot house, 1 square foot per 35 cubic feet.
Officers' mess, 1 square foot per 51 cubic feet.

When both heating and ventilating are required, the weight and space necessary to install a unit for each is very great. In order to interchange the system and to save weight, the Thermofan System may be installed. Air from the fan is discharged through, or drawn through a nest of pipes heated by steam. The temperature of the air is dependent upon the steam supply and may be varied at will. To insure that the air may not have too much of a drying effect upon the skin, jets of finely-divided water or steam are injected into the air. The system is flexible, as it may be used for either warming or cooling a compartment. For cooling, brine is circulated through the pipes. Tests have shown that heating by the Thermofan System is about four times as efficient as by direct steam-pipe radiation. The rise of temperature for a certain interval of time was four times as great as in the case of the steam pipe or radiator heaters under the same conditions of testing.

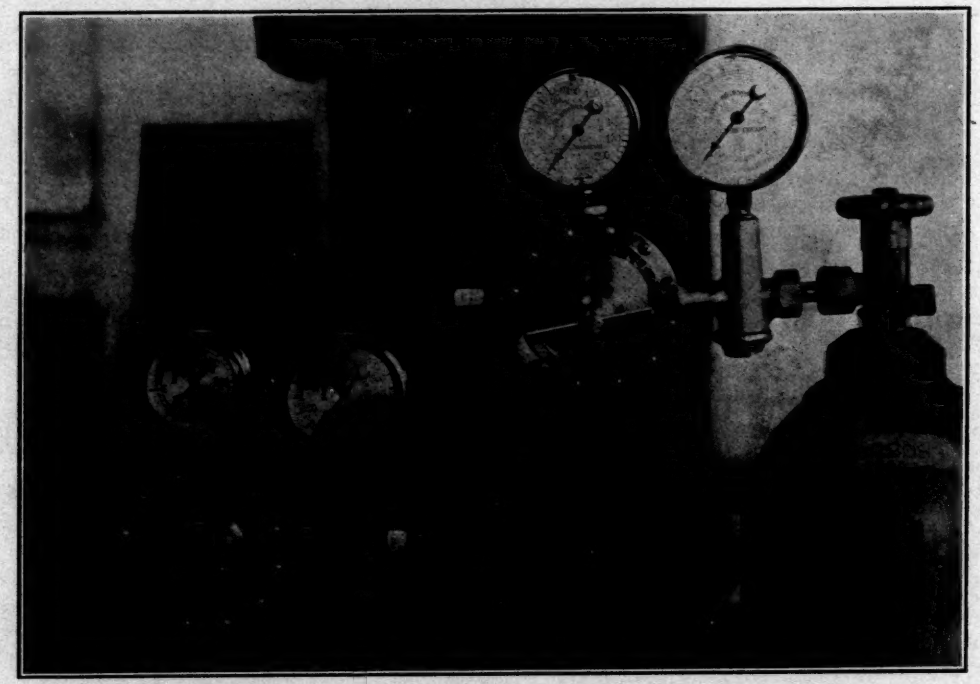


FIG. 1.

OXY-ACETYLENE WELDING.

By Stuart Plumley.*

The Oxy-Acetylene Process as applied to the welding of most metals and the cutting of iron and steel has become universal in recent years in connection with almost every industry in which metals are used.

Gases.—For welding purposes, oxygen and acetylene are almost exclusively used.

For brazing, lead burning, etc., oxygen and hydrogen or oxygen and illuminating gas or natural gas are used to a large extent.

For cutting, oxygen and acetylene are used to a greater extent than any other combination of gases, although for extremely heavy cutting, oxygen and hydrogen have been found advantageous. For all general cutting, the oxy-acetylene torch is preferable on account of its economy in consumption of oxygen and the rapidity with which the cut may be started.

Oxygen is produced to an enormous extent in the United States by two principal processes—the liquefaction of air and the subsequent separation of the oxygen from the nitrogen and the electrolysis of water. In the liquid air process the atmospheric air is reduced to liquefaction through pressure, refrigeration and expansion; in the liquid state and under pressure of a few pounds nitrogen, which liquefies at a lower temperature than that of oxygen, will evaporate first, and pass out of the machine. The oxygen is collected in gas form in gasometers, and is afterwards compressed in oxygen cylinders to a pressure of approximately 1,800 pounds per square inch. These cylinders are in daily use throughout the United States and may be obtained already charged from central distributing points located in almost every city.

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The second process consists in the electrolysis of water. By this means the electric current is utilized in an electrolytic cell to break up water into its component parts, oxygen and hydrogen.

Acetylene is usually generated in an acetylene generator. This generator contains calcium carbide in a hopper at the top of the machine and automatically feeds it into the water in the lower part of the apparatus. The reaction between the carbide and the water produces acetylene gas, which is piped directly to the point where it is used in the welding shop. Acetylene may thus be produced locally in any quantity required, and at a low operating cost.

Acetylene may also be procured, compressed and dissolved in cylinders. These cylinders are charged at central producing points and shipped to the user as required. The cylinder is a special cylinder completely filled with a porous filler of cement, charcoal, infusorial earth, etc., the filler being saturated with acetone. Acetone has the property of dissolving acetylene and thus making it safe while under pressure. Acetylene tanks are especially useful when the welding or cutting to be done necessitates the use of portable apparatus.

Pressure regulators or pressure reducers are used to reduce the high pressure of oxygen in the cylinders to suitable lower pressures at the torch, and the much lower pressures of acetylene flowing from the acetylene cylinders (or from the acetylene generator, if in use). Pressure gages indicate the pressures in the cylinders and at the torch. Regulators should always receive careful treatment. They are a senstive diaphragm mechanism, and the service is severe. If properly used they will accurately regulate the gas pressure. Fig. 1 shows the oxygen and acetylene regulators with the respective high-pressure and working-pressure gages.

The Oxy-acetylene Burner, known as the "torch," is connected to the source of supply by two lengths of hose. It is simple in construction. Fig. 2 shows how the oxygen and acetylene are brought together and intimately mixed in an

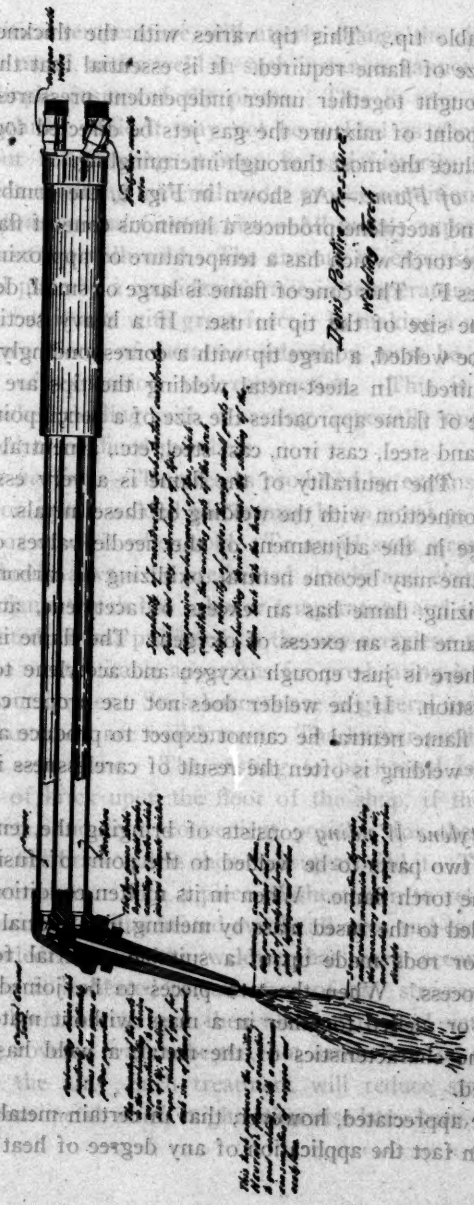


FIG. 2.—WELDING TORCH.

interchangeable tip. This tip varies with the thickness of weld and size of flame required. It is essential that the two gases be brought together under independent pressures, and that at the point of mixture the gas jets be directed together so as to produce the most thorough intermingling.

Character of Flame.—As shown in Fig. 2, the combustion of oxygen and acetylene produces a luminous cone of flame at the tip of the torch which has a temperature of approximately 6,300 degrees F. This cone of flame is large or small, depending upon the size of the tip in use. If a heavy section of metal is to be welded, a large tip with a correspondingly large cone is required. In sheet-metal welding the tips are small and the cone of flame approaches the size of a pencil point.

For iron and steel, cast iron, cast steel, etc., a neutral flame is required. The neutrality of the flame is a very essential feature in connection with the welding of these metals. By a slight change in the adjustment of the needle valves of the torch the flame may become neutral, oxidizing or carbonizing. The carbonizing flame has an excess of acetylene, and the oxidizing flame has an excess of oxygen. The flame is neutral when there is just enough oxygen and acetylene to produce combustion. If the welder does not use proper care in keeping the flame neutral he cannot expect to produce a good weld. Poor welding is often the result of carelessness in this respect.

Oxy-Acetylene Welding consists of bringing the temperature of the two parts to be welded to the point of fusion by means of the torch flame. When in its molten condition new metal is added to the fused mass by melting in it special welding stocks or rods made up of a suitable material for the work in process. When the two pieces to be joined have been fused or melted together in a mass without materially changing the characteristics of the metal, a weld has been accomplished.

It will be appreciated, however, that in certain metals such fusion, or in fact the application of any degree of heat above

the critical temperature, will utterly change the characteristics of the metal, and a weld in such instances cannot be made by the simple fusion of the pieces. This is true of malleable iron which, while it may not be welded as above described without re-annealing, it may be satisfactorily brazed with brass as a joining material under proper conditions.

Expansion and Contraction.—All metals expand with heat and contract with cold. The amount of expansion caused in a steel plate by a considerable rise in temperature is relatively small, but it acts with great force. In making a weld between any two pieces of metal consideration must be given to the effect of expansion and contraction. This is particularly true when dealing with a casting, especially one having restricted or confined members.

Pre-heating.—The stresses produced by expansion and contraction in the welded pieces may be so great as to break the weld or crack it when cold. To avoid such stresses the casting should always be pre-heated slowly and evenly by means of a flame which has a lower temperature and is not so concentrated as that produced by the oxy-acetylene torch.

The most suitable apparatus for pre-heating is a fuel oil or kerosene burner. Such burners are in general use in mechanical plants and are well-known. The burner produces a large, long, soft flame. The casting to be heated is set upon a layer of brick upon the floor of the shop, if the floor is of concrete, and in a convenient position for the welder. A rough fire-brick wall is then built up around it. The brick wall is covered over with a piece of sheet iron to retain the heat. The casting is thus heated evenly all over and each member is expanded. After the welding has been accomplished the casting is re-heated to take care of any stresses which may have been set up, and then allowed to cool very slowly by being covered with asbestos, hot sand, lime or ashes, to exclude the air. Such treatment will reduce stresses in the adjacent metal which might otherwise have been set up.

If the fuel oil pre-heaters are not obtainable charcoal may be used, preferably with a little compressed air. If neither of these can be obtained the casting may be set up on the fire brick so that the air will reach the charcoal fire. If the casting is extremely large it may be preferable not to heat it in its entirety, but any portion of it may be heated by building up that portion as described above. Large castings, however, require considerable skill in this respect so that preheating may be applied to the right members.

Size of Flame.—The size of the flame is governed by the amount of gas passing through the ports, and this flow of gas is controlled by the size of the interchangeable tip. The tip should be chosen for the thickness of metal. Sheet metal

TABLE I.

Davis-Bournonville Co. Style "C" Welding Torches.

Approximate Acetylene and Oxygen Pressures with Styles 99 and 100 Tips.

Tip No.	Thickness of Metal.	Acetylene Pressure.	Oxygen Pressure.
00	Very Light	1 lb.	1 lb.
0		1 lb.	2 lbs.
1	1/32–1/16 in.	1 lb.	2 lbs.
2	1/16–3/32 in.	2 lbs.	4 lbs.
3	3/32–1/8 in.	3 lbs.	6 lbs.
4	1/8–3/16 in.	4 lbs.	8 lbs.
5	1/4–5/16 in.	5 lbs.	10 lbs.
6	5/16–3/8 in.	6 lbs.	12 lbs.
7	7/16–1/2 in.	6 lbs.	14 lbs.
8	1/2–5/8 in.	6 lbs.	16 lbs.
9	5/8–3/4 in.	6 lbs.	18 lbs.
10	3/4–Up	6 lbs.	20 lbs.
11	Extra Heavy	8 lbs.	22 lbs.
12		8 lbs.	24 lbs.

For convenience, operators frequently adjust regulator to 2 pounds above that shown in table for the size of tip being used, which will allow for adjusting for drop caused by lowering of pressure in tank, variations in gauges, or other local causes.

will require a small flame and a small tip, while heavy plates require larger flames and larger tips. The size of the flame must, in short, bear a certain relation to the size of the work in order to obtain best results. Larger plates absorb and dissipate the heat rapidly, and if insufficient gas is burned to raise the joint to the fusing temperature welding cannot be accomplished. Absorption of heat is reduced by preheating, and rapid dissipation of heat is prevented by protecting the parts with some heat insulator like sheet asbestos, fire brick, etc.

Preparing the Joint.—If the parts to be welded are thin, little or no special preparation of the joint may be required, but on heavy parts it is necessary to chip away the edges to an angle of not less than 90 degrees. This may be accomplished with an air chipper, hand chisel or a grinding wheel, and is necessary in order that fusion of the joint may begin at its lower surface. When the torch is applied to the bottom of the V the lower edges are fused, and the sides of the parts are slowly raised to a fusing temperature.

When fusion has been accomplished throughout the immediate part of the section being welded, adding material is supplied by means of a suitable welding rod. It will, however, be appreciated that unless the joint is chipped away to permit the welder to begin at the bottom of the part, he cannot expect to create fusion clear through the section.

Adding Material (Welding Rods).—Adding material is usually made up in the form of wire of various diameters and approximately in 36-inch lengths, or in cast form of various diameters and approximately in 20-inch lengths. These rods, which are generally called welding rods, must be of a suitable alloy for the work in hand.

Most manufacturers, through a long series of experiments, have developed that certain alloys combined with certain fluxes produce the best results under given conditions, and it is essential that the welder keep this in mind in selecting a suitable welding rod for the job to be accomplished.

While these rods are usually made up upon a special analysis or formula proven best by experimental work, it is also necessary that the rods be made up by a special method and under definite conditions in order to produce satisfactory results. Two rods of like analysis will not always produce the same result when used for welding, unless the method of manufacture has been the same.

Manipulation of Torch.—Select a suitable size of tip for the work to be accomplished. This is generally shown in the manufacturer's pamphlets. Connect the torch to the source of supply of oxygen and acetylene, also as described in the manufacturer's directions. Open the valves, adjust the pressures and light the torch as indicated for the particular torch in use.

It will be noted that when the pressures of each gas are properly adjusted the flame will possess a clearly defined luminous cone. If a little more acetylene is turned into the torch a greenish tinge of flame will appear at the tip of the luminous cone. If a little more oxygen is turned on the cone will become shorter and more transparent and less luminous. To be neutral the flame should be just between these points. There should be no greenish tinge, but the flame should be clearly defined and luminous. See Fig. 2 for outline appearance of flame.

It is assumed that the work has already been prepared or preheated as has been described heretofore. Hold the torch in the right hand and the adding material in the left. Play the flame slowly along the edges to be united, beginning always at the edges farthest away from the operator. Play the torch momentarily at this point and quickly fuse the sides of the V until a puddle is formed. Apply the adding material in actual contact with the puddle and oscillate the torch slowly backward and forward in a semi-circle, keeping the flame and the tip pointing to the completed work. Oscillate the welding rod in similar manner to the torch, but in the opposite direction, which will keep the puddle free from

inclusions of gas. Continue this motion and keep the puddle flowing constantly until the weld is complete or until the operator has reached the point where he can permit the work to cool down. Do not permit the hot molten puddle to flow over the cold unmolten surface. If this is done, fusion between the puddle and the cooler surface will not occur, and a lap will be produced. Apply the torch in such manner that the parts being welded are always in a state of fusion before melting the welding rod into the puddle. See Fig. 3.

Fluxes.—The effect of a flux is to lessen oxidation and reduce the oxide film that prevents perfect union. The choice of flux is important, as success may hinge on the flux used. Home-made fluxes generally lack the careful proportioning of ingredients and the thorough grinding and mixing necessary for best results.

Direction of Welding.—When welding thin steel it is customary to tack at the left and weld from the right. But when thick sections are welded the welding should proceed from the left, as shown in Fig. 3. The advancing puddle and side walls are thus kept directly under the flame, and lapping and cold shuts are more easily avoided.

Welding Cast Iron.—Cast iron should always be preheated as described in the paragraph headed "Preheating." The edges of the crack or break should be carefully prepared before welding, by chipping away the metal so that there will be ample room to permit the welder to start fusion at the bottom of the groove. The casting should be placed in position so that its parts are in proper alignment and will remain so during welding. It is preferable to have the crack or break to be welded on the upper side of the casting.

When this has been accomplished the casting may be bricked up, using a rough firebrick wall, and preheated as above described. When the casting has reached a temperature a little less than dull red, the brick or cover may be removed to permit the welder to work.

In welding cast iron a special cast-iron welding rod should

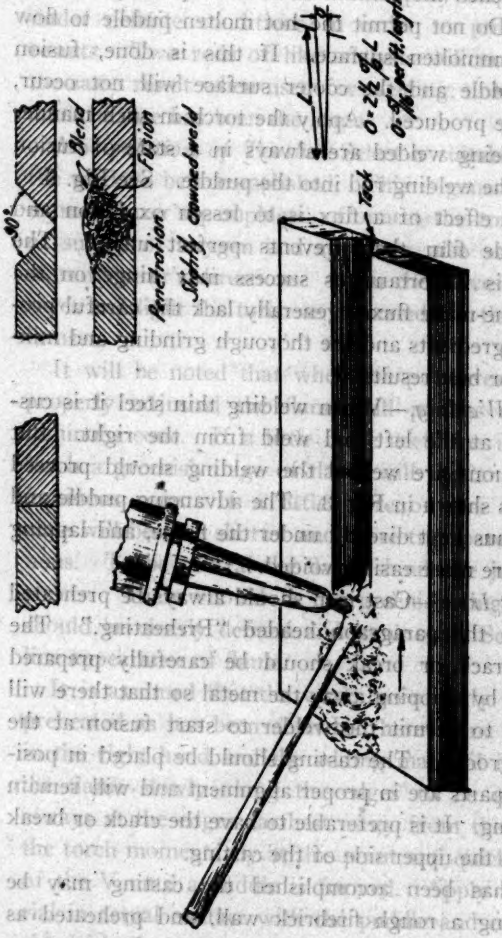


FIG. 3.—MANIPULATING THE TORCH.

always be used together with a proper flux. It is essential that the welding rod be high in silicon and of good grade of cast iron, and the flux properly prepared to keep the metal flowing smoothly.

As the sides of the V-break are fused and the added material united with the molten mass the welder uses the rod to scrape away any dirt or slag that may be encountered and to break up inclusions of gas. When the weld is completed in this manner the casting should be covered up and the preheating burners re-lighted for a short period of time to bring the heat again sufficiently high to eliminate any strain or stresses which may have been set up. The casting should then be completely covered up and allowed to cool very slowly.

Welding Malleable Iron.—Welding malleable iron is not advisable except where high strength of the joint is not required. The effect of the process by which white cast iron is converted into so-called malleable iron is destroyed by fusing the metal, as it is changed again to brittle cast iron. Broken malleable parts may be strongly united by brazing with bronze welding rods and flux. Care should be observed that too high a heat is not used in brazing. A full red heat produced by a neutral flame will yield best results. The iron and bronze should freely alloy in order to give a strong joint.

Welding Steel.—In welding steel, the welding rod used consists in most instances of a good grade of domestic iron wire, liquor drawn and annealed. In some instances it is preferable to use a low-carbon steel as a welding rod, but this depends upon the immediate case and the local conditions.

The plates should be prepared by chipping the edges to an angle of 90 degrees, and fusion should be produced at the bottom of the V and along its sides, before any welding wire is added. If the welder is not careful in this respect a series of cold-shuts and laps will be produced. Care must be taken not to burn the welding rod, and the puddle produced must be continually maintained as far as is practicable. The speed of rotation of the flame as it is manipulated by the welder, and

its advance along the seam, must be suited to the work in hand.

Welding Steel Castings.—Steel castings vary widely in carbon content, and no fixed rule can be given for welding. When the carbon content is below twenty points (0.20), welding can be accomplished with ease, but castings with higher percentage of carbon are likely to give trouble. A little copper added to the weld will assist in difficult cases, but its use in general is to be avoided, as it hardens the weld.

Welding Cast Aluminum.—The aluminum parts to be welded should be thoroughly cleaned to remove all grease, oxide and foreign material. The edges should be beveled the same as cast iron and steel if the thickness exceeds ¼ inch. Preheating is highly important in the case of aluminum, on account of its rapid heat absorption and dissipation. To secure perfect welds it is necessary to preheat comparatively small parts all over to a temperature of about 600 degrees F. Adding material is supplied in alloy sticks which should be free from sand or other foreign matter. A flattened iron rod is required to puddle the melted aluminum, break up the oxide and make it fill the joint. Care should be taken not to heat the end of the rod to a high temperature, as the resulting iron oxide will mix with the melted aluminum and cause a defective weld.

Welding Sheet Aluminum.—Sheet aluminum is welded with aluminum flux and drawn aluminum wire adding material. Care must be taken to choose a size of tip suited to the work. The operator must work rapidly, as thin sheets heat to the fusing point quickly, and the welding must be accomplished without delay in order to obtain a smooth, clean job.

Welding Copper.—Copper welding requires a larger tip and flame than steel of the same gage, because of the greater heat radiation, and the preheating of comparatively large pieces is necessary. It is desirable to use a flux and a strictly neutral flame to prevent oxidation and consequent weakening of the joint copper. A special alloy welding rod is required.

The joint copper has the tensile strength of cast copper, but it may be increased by judicious hammering.

Welding Brass and Bronze.—Care should be exercised in welding brass and bronze, not to overheat the metal and volatilize the zinc or tin. The white fumes resulting from overheating with the torch are poisonous, and the operator should avoid breathing them. A flux should be used, and preheating will be necessary on large pieces. Use special cast-bronze welding rod for adding material.

Welding Cast Iron and Steel.—The steel must first be heated to the welding temperature, as its fusing point is higher than that of cast iron. The torch flame is then directed against the cast iron, and, as fusion begins, adding material is supplied from regular cast-iron welding rods or sticks. The parts to be welded should be prepared the same as described for other metals. A cast-iron welding flux will facilitate the work, and is recommended.

Welding Copper and Steel.—Heat the steel to the welding point and apply the copper, letting it absorb heat from the steel until it begins to melt. The flame should be withdrawn as soon as fusion starts.

Welding High-Speed Steel and Machine Steel.—High-speed steel may be used economically for lathe and planer tools by welding cutting bits to machine steel shanks. The high-speed steel piece should be fitted to a notch in the end of the shank, which will afford support beneath. The sides of the bit to be welded should be heavily coated or "tinned" with regular welding rod, and when this is accomplished the welding to the shank is done in the usual way. The welding of high-speed drills to machine-steel shanks by the oxy-acetylene process can be done successfully, but is not advised unless special facilities are provided. If necessary for special work to provide a few long-shank high-speed drills, scarf the shanks and drills, and unite by brazing, using bronze welding rod and flux.

Welding Dissimilar Metals in General.—Metals having widely different melting temperatures may be successfully united by coating or "tinning" the metal having the higher melting point, with the metal having the lower melting point. Then the two are fused together. The flux to be used depends on the metals to be welded.

Cutting Iron and Steel.—A special type of oxy-acetylene torch is made for cutting wrought iron and steel. Cast iron

TABLE II.

Davis-Bournonville Co., Style "C" Cutting Torches.

Approximate Acetylene and Oxygen Pressures with Style 12 Tips.

Tip No.	Thickness of Metal.	Acetylene Pressure.	Oxygen Pressure.
1	1/8 in.	3 lbs.	10 lbs.
1	3/16 in.	3 lbs.	15 lbs.
1	1/4 in.	3 lbs.	20 lbs.
1	5/16 in.	3 lbs.	20 lbs.
2	1/4 in.	3 lbs.	10 lbs.
2	1/2 in.	3 lbs.	20 lbs.
2	3/4 in.	3 lbs.	30 lbs.
2	1 in.	3 lbs.	35 lbs.
3	1 in.	4 lbs.	30 lbs.
3	1 1/2 in.	4 lbs.	40 lbs.
3	2 in.	4 lbs.	50 lbs.
3	3 in.	4 lbs.	60 lbs.
4	3 in.	5 lbs.	60 lbs.
4	4 in.	5 lbs.	70 lbs.
4	5 in.	5 lbs.	85 lbs.
4	6 in.	5 lbs.	100 lbs.
5	6 in.	6 lbs.	90 lbs.
5	7 in.	6 lbs.	100 lbs.
5	8 in.	6 lbs.	125 lbs.
5	10 in.	8 lbs.	150 lbs.

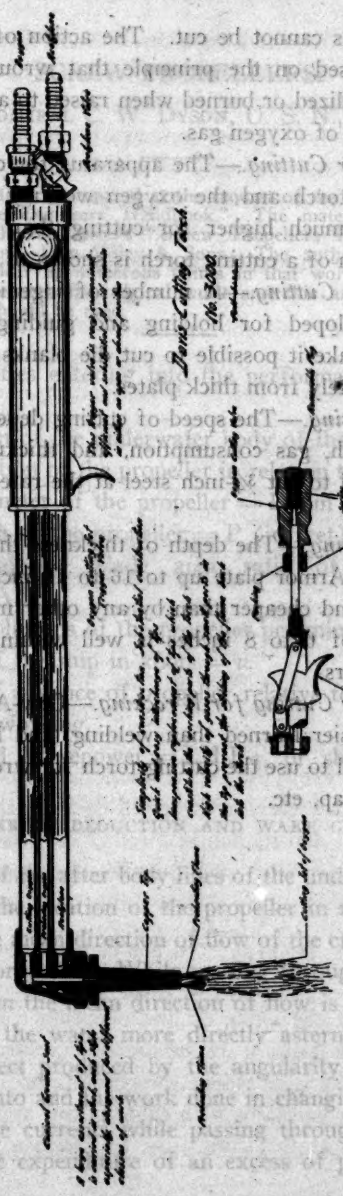


FIG. 4.—CUTTING TORCH.

and other metals cannot be cut. The action of cutting iron and steel is based on the principle that wrought iron and steel will be oxidized or burned when raised to a red heat and exposed to a jet of oxygen gas.

Apparatus for Cutting.—The apparatus for cutting differs in the type of torch and the oxygen working pressure, the pressure being much higher for cutting than for welding. The construction of a cutting torch is shown in Fig. 4.

Machines for Cutting.—A number of ingenious machines have been developed for holding and guiding the cutting torch. They make it possible to cut die blanks and all sorts of forms accurately from thick plates.

Speed of Cutting.—The speed of cutting depends upon the size of the torch, gas consumption, and thickness of steel. It is not unusual to cut ¾-inch steel at the rate of 15 inches per minute.

Depth of Cutting.—The depth of thickness that can be cut is remarkable. Armor plate up to 16 to 18 inches thick can be cut quicker and cheaper than by any other method. Cutting to depths of 6 to 8 inches is well within the skill of ordinary operators.

Oxy-Acetylene Cutting for Wrecking.—Oxy-Acetylene cutting is much easier learned than welding, and laborers may be quickly trained to use the cutting torch for wrecking operations, cutting scrap, etc.

SCREW PROPELLERS.

REAR ADMIRAL C. W. DYSON, U. S. N., MEMBER.

This article was first prepared at the request of the Editor as a section for the "Marine Engineers' Handbook." The material has been extracted from Admiral Dyson's "Screw Propellers for Hydraulic and Aerial Propulsion"—John Wiley & Sons. The author has taken this opportunity to clear up numerous points in that work, and where differences occur between the original work and this article, the material here presented should be used.

The Quantities entering into the performance of a screw propeller are:

1. The form of the underwater body of the ship.
2. The location of the propeller in relation to the hull.
3. The diameter of the propeller = D (in feet).
4. The pitch of the propeller = P (in feet).
5. The projected blade area ratio of the propeller = P.A. ÷ D.A.
6. The revolutions of the propeller per minute = R_d.
7. Speed of the ship in knots = v.
8. Speed of advance of propeller relative to water at stern in which it is working.
9. Indicated horsepower = I.H.P.$_d$, or shaft horsepower = S.H.P.$_d$

THRUST DEDUCTION AND WAKE GAIN.

The form of the after body lines of the underwater body of the ship and the location of the propeller in relation to these lines affect the mean direction of flow of the currents of water entering the propeller. While passing through the propeller some change in the mean direction of flow is produced, tending to throw the water more directly astern. The reduced propelling effect produced by the angularity of entrance of the currents into and the work done in changing the direction of flow of the currents while passing through the propeller necessitate the expenditure of an excess of power over that

which would be required were the lines of flow of these currents normal to the disc of the propeller. This excess expenditure of power is known under the name of *Thrust Deduction.*

Thrust Deduction is often stated to be the loss of propelling effect caused by the reduction of pressure of the water in the wake of the hull where the propellers are located. This is incorrect, for if this were the cause of the loss, the power required per revolution of the propeller would be reduced below that required where no loss existed. This, however, is not the case, as the same formula for apparent slip is used when the loss exists as when it does not, the apparent slip in both cases depending, among other factors, directly upon the power applied to the propeller, but not upon the fore-and-aft thrust delivered by the propeller.

Wake Gain.—As a vessel moves forward through the water she displaces the water which again flows in behind her. Some of this replacement flow occurs from forward, but not all of it, as the hull of the vessel prevents the full supply from being provided in that direction. The remainder of the flow comes from aft and results in reducing the relative velocity of hull to water below that between hull and distant water. This decrease in relative velocity is known as *Wake.* The actual slip of the propeller which works in this wake is known as the *real* slip, while the slip as measured between the speed of advance of the screw as it would be if working in a solid non-yielding nut and its actual speed of advance is known as the *apparent* slip. The ratio between the apparent slip and the *real* slip may be expressed as follows:

$$\frac{\text{Apparent slip}}{\text{Real slip}} = 1 - \text{Wake (in per cent of speed of advance of vessel)}.$$

The effect of the wake on the propeller varies in degree according to the position of the propeller relative to the hull. The general effect is to reduce the revolutions of the propeller below what they would be with equal power applied should no wake exist and at the same time to increase the thrust de-

livered by the propeller, thus tending to nullify some of the malign effect of thrust deduction. The closer in to the after body of the hull and the farther aft within the limits of the hull underwater body length the propeller is located, the greater will be the thrust deduction loss, but the greater will be the wake and the consequent wake gain. Calling the power required to drive a vessel without either *thrust deduction* or *wake gain*, unity, and using t to represent *thrust deduction* and w to represent *wake gain*, the factor by which unity power must be multiplied on account of the existence of these loss and gain factors is represented by

$$K = 1 + t - w.$$

SLIP BLOCK COEFFICIENTS.

Coefficients of the Hull.—The hull coefficients in common use for the comparison of hulls of vessels are four in number:

1. *Block Coefficient* = (Displacement in tons × 35) ÷ (Length on Load-water-line × Greatest immersed beam × Mean draught), all of these latter dimensions being in feet.
2. *Prismatic Coefficient* = (Displacement in tons × 35) ÷ {Area of immersed midship section (square feet) × Length on Load-water-line (in feet)}.
3. *Longitudinal Coefficient* = (Displacement in tons × 35) ÷ {Area Load-water-plane (square feet) × Mean draught (in feet)}.
4. *Midship Section Coefficient* = {Area of Immersed Midship Section (in square feet)} ÷ {Greatest immersed beam (in feet) × Mean draught (in feet)}.

Thrust deduction, wake gain, and *revolutions* of the propeller for any given power of engine and any speed of vessel depend upon the after body lines of the ship under consideration, and therefore upon the above coefficients. It would be preferable, however, to use the coefficients for the after body of the ship only, if these were available, but this is not the case at the present day. In determining approximately the

lines of the ship which affect the propeller performance, the *Block Coefficient* and *Midship Section Coefficient* are used.

If for a given length of middle body of the ship the after-body lines were of constant form and the midship section coefficient were constant for each particular *Block Coefficient*, it would be a simple matter to prepare charts from which the standard block coefficient could be picked off, as by holding the two given factors constant the length of the ship can be increased or decreased by varying the length of the middle body and yet the conditions of flow of the water to the propeller remain constant. The *Block Coefficient* of the ship would, however, increase or decrease with the length of the middle body and very nearly as the Beam (in feet) ÷ Length on Load-water-line (in feet).

The trials of such a standard set of vessels have been used in obtaining the factors for use in the design of propellers, and the hull coefficients of Block and Midship Section used in laying down a system by which the Standard (Slip) Block Coefficient of hulls of ordinary orthodox form obtained.

The *Equations* for estimates of Standard Slip Block Coefficients for various conditions are as follows:

WING SCREWS.

Class 1.—Propellers located comparatively close to the hull and the hull of such form that a vertical line through the center of the propeller pierces the skin of the ship at or below the water line, also where propellers are of small diameter and located well forward of the stern post, the vertical through the center of hub piercing the skin of the ship at or below the water line.

Formula A. Slip Block Coefficient (S.B.C.) $=$

$$\frac{\text{Beam} \div \text{Length on Load Water Line})}{.28\,(1 - \text{Block Coef.}) + (\text{Beam} \div \text{Length on Load Water Line})} =$$

$$\frac{B \div \text{L.L.W.L.}}{.28\,(1 - \text{B.C.}) + (B \div \text{L.L.W.L.})}$$

or, for close approximation to the above, Length between Perpendiculars, L.B.P., may be substituted for L.L.W.L., then,

$$\text{S.B.C.} = \frac{\text{B} \div \text{L.B.P.}}{.28\,(1 - \text{B.C.}) + (\text{B} \div \text{L.B.P.})}.$$

Class 2.—Where the after body of the hull fines so rapidly in the athwartship dimension that the vertical through the hub center passes entirely clear of the ship or pierces the hull well above the water line, the diameter of the propeller being greater than .75 × normal draught, the equation is, as before

$$\text{(A) S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.28\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})} = \frac{\text{B} \div \text{L.B.P.}}{.28\,(1 - \text{B.C.}) + (\text{B} \div \text{L.B.P.})}.$$

Class 3.—Where the hull is as in (1), but the propellers are small in diameter and located aft in way of stern post, the vertical through center of hub piercing the submerged hull as in (1), the equation becomes

$$\text{(B) S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.248\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})} = \frac{\text{B} \div \text{L.B.P.}}{.248\,(1 - \text{B.C.}) + (\text{B} \div \text{L.B.P.})}.$$

Class 4.—Where the after body is as in (2), but the propellers are located aft and close to the stern post and are of a diameter less than .7 normal draught of the ship, the propeller being located very low so that its lower blades pass close to or below the base line of the ship, the equation becomes

$$\text{(C) S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.567\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})} = \frac{\text{B} \div \text{L.B.P.}}{.567\,(1 - \text{B.C.}) + (\text{B} \div \text{L.B.P.})}.$$

Class 5.—Where the after body of the hull fines very rapidly in the vertical, but very slowly in the water line plane, the

vertical through the hub centers piercing the hull of the vessel well above the water line, the equation becomes

$$(1)\ (A)\ \text{S.B.C.} = \frac{B \div \text{L.L.W.L.}}{.28\ (1 - \text{B.C.}) + (B \div \text{L.L.W.L.})} = \frac{B \div \text{L.B.P.}}{.28\ (1 - \text{B.C.}) + (B \div \text{L.B.P.})}.$$

Where the shafts are carried in spectacle frames and the propellers are far aft and close to the stern post,

$$(2)\ (D)\ \text{S.B.C.} = \frac{B \div \text{L.L.W.L.}}{.2\ (1 - \text{B.C.}) + (B \div \text{L.L.W.L.})} = \frac{B \div \text{L.B.P.}}{.2\ (1 - \text{B.C.}) + (B \div \text{L.B.P.})}.$$

SINGLE SCREW SHIPS.

Class 6.—Where the hull is of such form that a vertical through the hub center pierces the hull well above the water line, the hull being of not less than 20 feet draught at the stern and the propeller not less than .75 × draught in diameter, or (2) where the hull is of such form that the propeller is completely shrouded by the load water plane of the ship, the equation for the Slip Block Coefficient is

$$(E)\ \text{S.B.C.} = \frac{B \div \text{L.L.W.L.}}{.164\ (1 - \text{B.C.}) + (B \div \text{L.L.W.L.})} = \frac{B \div \text{L.B.P.}}{.164\ (1 - \text{B.C.}) + (B \div \text{L.B.P.})}.$$

Class 7.—Where the hull is of such form that the vertical through the hull center pierces the hull well above the water line and where the draught at the stern is less than 14 feet or the diameter of the propeller is less than .7 the draught at the stern, the equation takes the form

$$(C)\ \text{S.B.C.} = \frac{B \div \text{L.L.W.L.}}{.567\ (1 - \text{B.C.}) + (B \div \text{L.L.W.L.})} = \frac{B \div \text{L.B.P.}}{.567\ (1 - \text{B.C.}) + (B \div \text{L.B.P.})}.$$

FOR SUBMARINES.

Class 8.—Single-hull type with propellers located abaft and clear of hull.

$$\text{(F) S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.064\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})} = \frac{\text{B} \div \text{L.B.P.}}{.064\,(1 - \text{B.C.}) + (\text{B} \div \text{L.B.P.})}.$$

Class 9.—Double-hull type with propellers located beneath hull.

$$\text{(G) S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.044\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})} = \frac{\text{B} \div \text{L.B.P.}}{.044\,(1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})}.$$

FOR TUNNEL BOATS.

Class 10.—The S.B.C. will be, at the greatest, that formed by using formula (A), and may be still less. Data on the performance of such vessels are very meager at the present time.

CORRECTION OF S.B.C. FOR VARIATION OF MIDSHIP SECTION COEFFICIENT FROM STANDARD.

The Standard Midship Section Coefficients agree with the values of the S.B.C. as found by equation (A). Should there be a wide variation between the actual and the standard, certain corrections should be made to conditions existing under Classes 1, 3, 4, 5, 6 and 7 to obtain the proper S.B.C.

The values of the Standard Midship Section Coefficients corresponding to the values of S.B.C. as obtained by equation (A) are

TABLE I.

S.B.C.	.95	.9	.85	.8	.75	.7	.65	.6	.55	.5	.45	.4	.35	.3
Mid. Sec. Coef.	.996	.994	.992	.99	.988	.985	.981	.978	.952	.892	.803	.712	.625	.535

To illustrate the method of correction for variation in midship section from standard, suppose the S.B.C. as found by equation (A) be .6, but the midship section coefficient be .95, then the corrected S.B.C. for the class 1 Condition will be

$$\text{S.B.C. (Class 1)} = .6 \times \frac{.978}{.95} = .608.$$

Now suppose the vessel be single-screw of Class 4, where

$$\text{(E)}\quad \text{S.B.C.} = \frac{\text{B} \div \text{L.L.W.L.}}{.164\ (1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})},$$

$$\text{(A)}\quad .608 = \frac{\text{B} \div \text{L.L.W.L.}}{.28\ (1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})},$$

$$\text{S.B.C.} = \frac{.608 \times .28\ (1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})}{.164\ (1 - \text{B.C.}) + (\text{B} \div \text{L.L.W.L.})}$$

which will be the corrected Slip Block Coefficient for the single screw vessel of this same form.

DETERMINATION OF THRUST DEDUCTION FACTOR.

As already stated, the Thrust Deduction Factor K for any hull depends upon the form of the after body of the hull, the Slip Block Coefficient, and upon the position of the propeller relative to the hull; thus the S.B.C. to use in determining the value of K for vessels of Classes as given under section on Slip Block Coefficient, are

Class 1, is that found by Equation (A) corrected for variation in midship section coefficient from Standard.

Class 2, is that found by Equation (A).

Class 3, is that found by Equation (B) corrected for variation in midship section coefficient from Standard.

Class 4, is that found by Equation (C) corrected for variation in midship section coefficient from Standard.

Class 5, (1), is that found by Equation (A) corrected for variation in midship section coefficient from Standard.

Class 5, (2), is that found by Equation (D) corrected for variation in midship section coefficient as before.

Class 6, (1) and (2), is that found by Equation (E) corrected as for class (5).

Class 7, is that found by Equation (E) corrected as for class 5.

Class 8, the value of K is unity.

Class 9, is unity when good hull clearance and good distance between propellers is maintained.

Class 10, is of constant value and K = 1.195.

MEAN RELATIVE TIP CLEARANCE OF PROPELLERS.

In vessels of Classes 1, 3, 4, 6 (2), and 9 the relative value of K is variable and is dependent upon the relative mean tip clearance between the propeller blade tip in closest position to the hull and the hull, the equation for finding the relative mean tip clearance being

$$\text{M.T.C.} = \frac{\text{H} + 10\text{H}_1}{11}, \text{ where}$$

H = (Actual immersion, in feet, of tip of upper blade of propeller when in the vertical position × 14) ÷ (Actual height of this tip above horizontal base line of ship, in feet, this base line being a line drawn tangent to the keel at its greatest immersion.)

H_1 = (Actual minimum tip clearance between circumference of propeller disc and hull × 10) ÷ (Actual height of center of hub of propeller, in feet, above the base line.)

The value of K for vessels of these classes for different values of S.B.C. as found above and for varying values of M.T.C. are given in Table II.

A good general rule to follow is to keep the tips of the blades as far from the hull body as possible and yet close enough in to the hull to retain the full value of the wake while at the same time the shaft lines are kept parallel to the direction of motion of the vessel so that the currents of water entering the propeller are given an opportunity to straighten

out and enter the propeller in a direction more nearly perpendicular to its disc, while the direction of thrust of the propeller will be as nearly opposite to the direction of motion of the vessel as possible.

S.B.C.	Relative Mean Tip Clearance in Feet						
	1.5	2.0	2.5	3.0	3.5	4.0	4.5
	Values of K						
.3	1.0	1.0	1.0	1.0	1.0	1.0	1.0
.35	1.0	1.0	1.0	1.0	1.0	1.0	1.0
.4	1.07	1.0	1.0	1.0	1.0	1.0	1.0
.45	1.093	1.03	1.0	1.0	1.0	1.0	1.0
.5	1.125	1.078	1.005	1.0	1.0	1.0	1.0
.55	1.175	1.10	1.0675	1.0	1.0	1.0	1.0
.6	1.25	1.14	1.09	1.064	1.01	1.0	1.0
.65	1.34	1.19	1.125	1.086	1.064	1.025	1.0
.7	1.45	1.275	1.183	1.115	1.085	1.066	1.04
.75	1.58	1.37	1.255	1.175	1.112		
.8	1.75	1.47	1.34	1.24			
.85	2.00	1.6	1.425				
.9	2.26	1.75					

For submarines of the Lake type on the surface, the value of K is as taken from this Table II. For the submerged condition, the loss will only be 1/2 as great, thus if K on the surface = 1.34, in the submerged condition it will equal 1.17.

Table II. Values of K for M.T.C. and S.B.C.

*With comparatively large tip clearances between propellers and hull and between adjacent propellers, K for this class vessel may safely be assumed as unity.

For shallow-draft vessels such as destroyers the actual tip clearance should not be less than two feet, while for vessels of deep draught where the propeller is overhung by the immersed body of the hull, the actual tip clearance should be not less than three feet, and more if possible.

The above Tables of K values are for Standard projected area ratio form blades. Where blades are of the forms known as *Fan, Broad Tipped*, or *Narrow Tipped*, as described later, these values as given for vessels in which the values of K

depend upon the relative mean tip clearance, may be considerably modified, increasing for the first two types and decreasing for the third type, the actual value for the first two forms being the mean between that for the actual tip clearance and that for the Basic propeller from which these other forms are deduced.

S.B.C.	.6	.625	.65	.675	.7	.725	.75
K	1.0	1.04	1.085	1.08	1.093	1.115	1.16

S.B.C.	.775	.8	.825	.85	.875	.9	.925
K	1.2	1.27	1.34	1.425	1.57	1.78	2.4

Table III. Values of K for Vessels of Classes 2 and 6(1).

S.B.C.	.4	.45	.5	.55	.6	.65	.7	.75
K	1.0	1.01	1.07	1.10	1.135	1.19	1.28	1.4

S.B.C.	.8	.85	.9
K	1.58	1.94	3.2

Table IV. Values of K for Vessels of Class 5.

S.B.C.	.55	.6	.65	.7	.75	.8	.85	.9
K	1.0	1.07	1.10	1.175	1.27	1.385	1.54	1.775

Table V. Values of K for Vessels of Class 7.

RESISTANCE OF HULL APPENDAGES.

The hull appendages usually fitted depend upon the character of the vessel and are generally as follows:

1. Battleships, multiple propellers:—Rudder, bilge keels, shaft struts and docking keels.

2. Merchant ships, multiple propellers:—Rudder, bilge keels, shaft struts or spectacle frames.

3. Fast and light vessels, multiple propellers:—Rudder, shaft struts and, in some cases, bilge keels and injection scoops.

4. Single-screw vessels:—Rudder and rudder post and, in large vessels, bilge keels.

D.K.-Docking Keel
B.K.-Bilge Keel
Rud.-Rudder
Sh.-Shaft
St.-Strut

$\frac{v}{\sqrt{L.W.L.}}$	B.K. & Rud.	Sh. & St. a B.K. & Rud.	Sh. & St. a Sh. & St. & B.K. & Rud.	Sh. & St. of Fine Section. B.K. & Rud.	Sh. & St. of Heavy Section. B.K. & Rud.	Destroyers Sh. & St. B.K. Rud. & Scoops.
.3	.024	.054	.083	.105	.128	.093
.4	.028	.065	.115	.129	.156	.115
.5	.034	.083	.146	.164	.195	.146
.6	.044	.118	.192	.216	.258	.192
.7	.059	.170	.260	.279	.312	.260
.8	.080	.139	.202	.224	.282	.202
.9	.021	.080	.133	.154	.213	.133
1.0	.008	.057	.098	.115	.153	.098
1.1	.01	.055	.094	.109	.133	.094
1.2	.02	.070	.117	.127	.143	.117
1.3	.028	.080	.120	.131	.147	.120
1.4	.019	.086	.104	.116	.132	.104
1.5	.004	.051	.083	.093	.104	.083
1.6	.004	.050	.080	.090	.100	.080
1.7*	.004	.050	.080	.090	.100	.080

*From This Point the Appendage Resistance Remains Constant, Probably Falling for Still Higher Values.

Table VI. Appendage Resistance in Per cent of Bare Hull for Battleships and Destroyers.

$v \div \sqrt{L.W.L.}$	.4	.5	.6	.7	.8	.9	1.0
App. Resis., %	.188	.138	.115	.105	.101	.10	.10

Table VII. Appendage Resistances in Per cent. of Bare Hull, Merchant Vessels.

The resistance of the appendages appears to be a varying percentage of the bare hull resistance, based on the values of v (Speed in knots) $\div \sqrt{\text{L.L.W.L.}}$

For vessels of lines similar to battleships and destroyers Table VI gives the approximate values of the appendage resistance in per cent of the bare hull resistance:—

For vessels of Merchant Type, given as Class 5 under discussion of S.B.C., where the appendages are usually Rudder, Struts, and Bilge Keels, the percentage appendage resistances appear to be as given in Table VII.

These percentages from either Table are additive to the bare hull resistance and must be applied where the total effective horsepower required to tow the vessel is given as the base on which to design the propeller and estimate the necessary power. Any abnormal appendages must be estimated independently and added to the total e.h.p. as above obtained.

BASIC CONDITION FOR ANALYSIS AND DESIGN OF SCREW PROPELLERS.

So long as the blade sections of propellers approximate closely to the standard form of blade section described later, all propellers, no matter what the form of projected area ratio, may be reduced to a standard form of this ratio, and the performance of this standard formed propeller be estimated when working under a given fixed condition of resistance, and by means of coefficients obtained from actual performances of many propellers behind many different varieties of hulls, the proper corrective factors to obtain the performances under any other conditions of resistance can be estimated and applied.

This condition of fixed resistance is known as the Basic Condition of Design and the Propellers and Data conforming to it are known as Basic. The Table of these conditions (Table VIII) gives for increasing values of Projected Area Ratios for three-bladed propellers, propellers of this number blades having been used in obtaining the data, the following values:

Table VIII. Basic Condition of Analysis and Design of Screw Propellers (Three Blades).

P.A./I.D. D.A.	I.H.P. lbs.	F.S. Ft.per P.O. Minute	%	Slip Block Coefficient									
				0	.1	.2	.3	.4	.5	.6	.7	.8	.9	1
.15	1.10	3850	69.9	.8	.809	.827	.85	.869	.889	.902	.939	.95	.97	.992
.175	1.45	3520	70.5	.8	.809	.827	.849	.869	.887	.902	.939	.948	.97	.992
.2	1.85	4250	70.5	.799	.809	.827	.848	.868	.884	.902	.936	.946	.969	.991
.225	2.28	4880	70.7	.796	.809	.824	.845	.867	.883	.901	.934	.946	.967	.990
.25	2.76	5500	70.1	.796	.808	.824	.843	.865	.881	.900	.932	.944	.964	.988
.275	3.25	6100	69.2	.794	.808	.823	.842	.863	.880	.896	.930	.940	.962	.986
.300	3.75	6650	68.2	.792	.807	.821	.841	.861	.878	.894	.918	.936	.958	.981
.325	4.27	7150	66.7	.789	.803	.820	.840	.860	.878	.894	.915	.935	.955	.978
.350	4.8	7660	65.2	.785	.801	.817	.838	.858	.874	.891	.912	.931	.951	.975
.375	5.4	8120	63.7	.781	.799	.814	.835	.854	.873	.891	.908	.931	.951	.975
.400	6.04	8600	61.9	.777	.796	.810	.831	.850	.870	.888	.904	.927	.947	.972
.425	6.65	9050	60.3	.773	.793	.806	.827	.846	.866	.884	.900	.923	.943	.969
.450	7.32	9500	58.6	.769	.788	.801	.823	.842	.861	.880	.896	.919	.939	.963
.475	8.03	10020	56.9	.763	.780	.796	.818	.837	.855	.875	.894	.913	.933	.956
.500	8.75	10550	55.4	.756	.775	.791	.811	.831	.849	.869	.888	.908	.927	.950
.525	9.50	11150	53.8	.748	.766	.784	.804	.824	.841	.860	.881	.901	.927	.950
.550	10.3	11850	52.6	.738	.758	.775	.796	.813	.832	.851	.873	.897	.917	.940
.575	11.1	12620	53.5	.726	.747	.765	.784	.802	.822	.840	.863	.885	.905	.928
.600	11.97	13530	53.5	.715	.735	.752	.771	.790	.807	.827	.848	.871	.870	.913
.625	12.83	14580	53.5	.700	.720	.736	.755	.772	.792	.810	.831	.851	.852	.895
.650	13.71	15780	52.5	.682	.700	.717	.735	.753	.772	.792	.810	.831	.830	.878

1. $I.T._D$ = Basic Indicated thrust in pounds per square inch of disc area, including the hub.

2. T. S. = Basic Tip speed of propeller in feet per minute.

3. $1 - S = 1 -$ Basic Apparent Slip for any S.B.C., the apparent slip being in per cent. As the conditions are fixed, the relation between the real slip and the apparent slip for any S.B.C., $I.T._D$ and T.S. will be constant and therefore the apparent slip may be substituted for the real slip in the computations.

4. P.C. = Basic Propulsive Coefficient in per cent.

(A) From the values given in this table, having given

P = Pitch of Propeller, in feet.

D = Diameter of Propeller, in feet.

$\frac{P.A.}{D.A.}$ = Projected Area Ratio of 3 Blades of the propeller.

$4/3\,\frac{P.A.}{D.A.}$ = Total Projected Area Ratio—4 blades.

$2/3\,\frac{P.A.}{D.A.}$ = Total Projected Area Ratio—2 blades.

The values

I.H.P. = Basic Indicated Horsepower.

E.H.P. = Basic Effective (Tow Rope) Horsepower.

S.H.P. = Basic Shaft Horsepower.

Can be obtained where

$$I.H.P. = \frac{D^2 \times I.T._D \times P \times R}{C}$$

$$C = \begin{cases} 252.4 \text{ for 4-bladed propeller.} \\ 291.8 \text{ for 3-bladed propeller.} \\ 389.0 \text{ for 2-bladed propeller.} \end{cases}$$

S.H.P. = I.H.P. × .92 for Standard Conditions.

E.H.P. = I.H.P. × P.C., where P.C. is the propulsive coefficient taken from Table VIII for the total projected area ratio, all other data being taken for P.A. ÷ D.A., no matter what the number of blades.

In addition may also be obtained the values

V = Basic speed of the ship,

R = Basic revolutions of the propeller.

Where

$$V = \frac{R \times P \times (1 - S)}{101.33}.$$

$$R = \frac{T.S.}{\pi D}.$$

(B) From this same Basic data as given in the Table, should it be desired to design the propeller for Basic conditions of resistance, and having given I.H.P. or S.H.P. and V, and assuming any one or several values of P.A. ÷ D.A., the values D, P, R, and P.C. can be obtained, as the value of P.C. for P.A. ÷ D.A. or for the total projected area ratio is fixed by the Table, while

$$P \times R = \frac{V \times 101.33}{1 - S}.$$

$$D = \sqrt{\frac{C \times I.H.P.}{I.T._D \times P \times R}}, \text{ C having same values as before.}$$

$$P = \frac{\pi \times D \times 101.33 \times V}{T.S. \times (1 - S)}.$$

$$R = \frac{T.S.}{\pi D}.$$

E.H.P. = I.H.P. × P.C. as before.

S.H.P. = I.H.P. × .92 as before.

Should the mechanical efficiency be greater or less than .92, say A, then the actual indicated horsepower required will not be I.H.P., but will be I.H.P. $\times \frac{.92}{A}$, while the E.H.P. and the S.H.P. will remain as before.

Should *Thrust Deduction* exist, the actual value of the Indicated Horsepower will be $K \times I.H.P.$, that of the shaft horsepower $K \times S.H.P.$, while that of E.H.P. will remain $I.H.P. \times P.C.$

The apparent slip in this latter case will become KS instead of S, and the revolutions will be

$$R_d = \frac{V \times 101.33}{P \times (1 - KS)}$$

(C) Should E.H.P. and V be given, the formulas take the following forms:

$I.H.P. = E.H.P \div P.C.$

K = Thrust Deduction Factor.

$K \times I.H.P.$ = Total Indicated Horsepower.

$K \times I.H.P. \times .92$ = Indicated Shaft Horsepower.

$K \times I.H.P. \times \frac{.92}{A}$ = Total Indicated Horsepower for engine mechanical efficiency differing from .92.

$$D = \sqrt{\frac{C_1 \times E.H.P.}{E.T._p \times \frac{P.A.}{D.A.}}}, \text{ where}$$

$$C_1 = \begin{cases} 2.491 \text{ for 4 bladed propellers.} \\ 2.88 \text{ for 3 bladed propellers.} \\ 3.84 \text{ for 2 bladed propellers.} \end{cases}$$

and

$$E.T._p = \frac{P.T._p}{1 - S}.$$

$$P.T._p = I.T._p \times P.C.$$

$$I.T._p = I.T._D \div \frac{P.A.}{D.A.}.$$

Should it be desired to use the value of I.H.P. as obtained above, the equation for diameter becomes

$$D = \sqrt{\frac{C_1 \times I.H.P. \times (1 - S)}{I.T._D \times V}}, \text{ where}$$

C_1 has the values given above.

P, R and R_a are found as before.

(D) Finally, in addition to I.H.P. or S.H.P. and V in (B) or E.H.P. and V in (C) there may also be assumed the value of D. In such cases the value of P.A. ÷ D.A. and the total value of projected area ratio will be found in addition to

	Values of P.A.÷D.A.									
S.B.C.	.3	.35	.3	.35	.4	.45	.5	.55	.6	.65
	Values of $I.T._p \div (1-S)$									
.9	1.93	2.87	3.89	5.04	6.31	7.80	9.37	11.3	13.43	15.57
.85	1.95	2.90	3.93	5.10	6.38	7.89	9.50	11.35	13.61	15.71
.8	1.97	2.93	3.97	5.16	6.45	7.96	9.58	11.45	13.74	15.94
.75	2.00	2.97	4.02	5.22	6.53	8.08	9.71	11.65	13.93	16.14
.7	2.02	3.00	4.07	5.29	6.61	8.18	9.85	11.79	14.11	16.30
.65	2.05	3.04	4.12	5.35	6.69	8.27	9.98	11.94	14.38	16.54
.6	2.07	3.08	4.17	5.42	6.77	8.37	10.07	12.08	14.47	16.75
.55	2.09	3.11	4.21	5.48	6.85	8.45	10.21	12.23	14.65	16.96
.5	2.11	3.14	4.26	5.53	6.90	8.55	10.30	12.36	14.81	17.18
.45	2.14	3.17	4.30	5.59	6.98	8.65	10.41	12.51	14.95	17.39
.4	2.16	3.20	4.35	5.65	7.06	8.75	10.53	12.65	15.15	17.59
.35	2.19	3.24	4.40	5.71	7.14	8.86	10.66	12.79	15.32	17.81
.3	2.21	3.29	4.45	5.79	7.23	8.96	10.79	12.95	15.52	18.05

Table IX. Values of $I.T._p \div (1-S)$ for Varying Values of P.A.÷D.A.

	Values of P.A.÷D.A.									
Slip B.C.	.2	.25	.3	.35	.4	.45	.5	.55	.6	.65
	Values of $(P.A.\div D.A.)\times E.T._p$									
.9	1.37	2.01	2.65	3.29	3.9	4.57	5.19	5.89	7.05	8.17
.85	1.38	2.04	2.68	3.32	3.95	4.62	5.25	5.98	7.15	8.25
.8	1.4	2.06	2.71	3.36	3.99	4.67	5.31	6.03	7.21	8.37
.75	1.42	2.09	2.74	3.40	4.04	4.73	5.37	6.13	7.31	8.47
.7	1.43	2.11	2.77	3.45	4.09	4.79	5.44	6.2	7.41	8.56
.65	1.45	2.14	2.81	3.49	4.14	4.88	5.51	6.28	7.50	8.68
.6	1.47	2.16	2.84	3.52	4.19	4.91	5.58	6.36	7.6	8.79
.55	1.485	2.19	2.87	3.57	4.24	4.95	5.64	6.43	7.69	8.91
.5	1.5	2.21	2.90	3.61	4.27	5.01	5.7	6.50	7.77	9.02
.45	1.518	2.23	2.93	3.65	4.32	5.07	5.76	6.58	7.85	9.13
.4	1.53	2.25	2.97	3.68	4.37	5.13	5.84	6.65	7.95	9.24
.35	1.55	2.28	3.00	3.72	4.42	5.19	5.91	6.73	8.04	9.35
.3	1.57	2.31	3.04	3.77	4.48	5.25	5.98	6.81	8.15	9.48

Table X. Values of $(P.A.\div D.A.)\times E.T._p$ for Varying Values of P.A.÷D.A.

P, R, R_a, and the power values sought. The equations become either

$$I.T._p \div (1-S) = \frac{C_1 \times I.H.P.}{D^2 \times V} \text{ for (B), or}$$

$$E.T._p \times \frac{P.A.}{D.A.} = \sqrt{\frac{C_1 \times E.H.P.}{D^2 \times V}} \text{ for (C), the values of}$$

$I.T._D \div (1 - S)$ for different values of P.A. ÷ D.A. being given in Table IX, while those for $E.T._p \times \frac{P.A.}{D.A.}$ for different values of P.A. ÷ D.A. are given in Table X.

CORRECTIVE FACTORS FOR POWER FOR OTHER THAN BASIC CONDITIONS OF DESIGN.

Suppose the Basic Conditions of any given propeller be

I.H.P. = Basic Indicated Horsepower,

E.H.P. = Basic Effective (tow rope) Horsepower,

S = Basic Apparent slip for S.B.C. of hull to which propeller was applied,

V = Basic Speed in knots.

Suppose further that this propeller working behind the given hull delivered a speed

v = Actual speed in knots.

$I.H.P._d$ = Actual Indicated horsepower,

A = Actual percentage mechanical efficiency of the engine,

K = Thrust deduction factor for type of hull and position of propeller,

$I.H.P._p = I.H.P._d \times \frac{A}{.92} \div K$ = Actual power delivering net effective horsepower in propelling the vessel.

e.h.p. = Net effective horsepower required for speed v.

e.h.p. = Gross effective horsepower corresponding to

$$I.H.P._d \times \frac{A}{.92}$$

(1) Being given $I.H.P._d$ to find e.h.p. and R_d, where R_d = Estimated Revolutions per minute. $10^* = I.H.P. \div I.H.P._p$,

e.h.p. ÷ E.H.P = Power percentage corresponding to Z from Table XII,

e.h.p. = E.H.P. × (e.h.p. ÷ E.H.P.) = net effective horsepower used in propulsion.

$10^2 = \text{I.H.P.} \div \left(\text{I.H.P.}_d \times \frac{A}{.92}\right)$,

e.h.p. ÷ E.H.P. = Power percentage corresponding to Z from Table XII,

e.h.p. = Gross effective horsepower for which water must be supplied to the propeller.

e.h.p. must never be so great that e.t. > 1.15 E. T., where

$$\text{e.t.} = \frac{e.h.p. \times 33{,}000}{v \times 101.33}.$$

while $\text{E.T.} = \dfrac{\text{E.H.P.} \times 33{,}000}{\text{V} \times 101.33}$.

This requirement is necessary to guard against cavitation of the suction to the propeller.

To guard against breaking down or dispersion of the thrust column, the values of e.t. where $\text{e.t.} = \dfrac{\text{e.h.p.} \times 33{,}000}{v \times 101.33}$, should never exceed certain values which values correspond with the values of $v \div \text{V}$ and e.h.p. ÷ E.H.P. given in Table XI.

In designing, the propeller should always, if possible, have such characteristics that the value of $v \div \text{V}$ for any load e.h.p. ÷ E.H.P. will be greater than the value of $v \div \text{V}$ given in Table XI for the same value of e.h.p. ÷ E.H.P.

CORRECTIVE EQUATION FOR REVOLUTIONS.

The revolutions to be expected with any power and speed with any hull, the propeller working under other than basic conditions of resistance, can be obtained as follows:

$$\text{Apparent slip under actual conditions} = s = \text{S} \times \frac{\text{I.H.P.}_d \times \text{V}^y}{\text{I.H.P.} \times v^y}.$$

Where I.H.P._d and v are the actual power and speed while I.H.P., V and S are the power, speed and apparent slip under basic conditions.

This equation may also be written in terms of shaft horsepower or in terms of K and Z, thus

$$s = \text{S} \frac{\text{S.H.P.}_d \times \text{V}^y}{\text{S.H.P.} \times v^y} = \text{S} \frac{\text{K} \times \text{V}^y}{10^z \times v^y},$$

but the latter form should only be used where there is no danger of cavitation or of dispersal of thrust columns.

V^y and v^y are written as A_v and A_v, and Table XIII gives the values of Log A_v in all cases, and of Log A_v in cases of vessels of Classes 1, 2, 3, 4, 5, 6, 7, 8 on the surfaces, and 10, these cases being those given under S.B.C.

$\frac{e.h.p.}{E.H.P.}$	.05	.075	.1	.15	.2	.25	.3	.35
$\frac{v}{V}$	.308	.36	.4	.46	.507	.545	.578	.605
$\frac{e.h.p.}{E.H.P.}$	.4	.45	.5	.55	.6	.65	.7	.75
$\frac{v}{V}$	.628	.65	.666	.68	.69	.7	.709	.716
$\frac{e.h.p.}{E.H.P.}$	.8	.85	.9	.95	1.0	1.05	1.10	1.15
$\frac{v}{V}$	.721	.74	.762	.825	.868	.912	.956	1.0
$\frac{e.h.p.}{E.H.P.}$	1.2	1.25	1.3	1.35	1.4	1.45	1.5	1.55
$\frac{v}{V}$	1.042	1.088	1.132	1.175	1.218	1.262	1.304	1.347
$\frac{e.h.p.}{E.H.P.}$	1.6	1.65	1.7	1.75	1.8	1.85	1.9	
$\frac{v}{V}$	1.392	1.435	1.478	1.522	1.567	1.61	1.652	

Table XI.

$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z	$\frac{e.h.p.}{E.H.P.}$	Z
.025	-1.668	.55	-.2704	1.15	+.0632	1.75	+.2531
.05	-1.385	.6	-.231	1.2	+.0825	1.80	+.2658
.075	-1.1715	.65	-.1948	1.25	+.1009	1.85	+.2782
.100	-1.0414	.7	-.1613	1.3	+.1186	1.9	+.2903
.15	- .858	.75	-.1301	1.35	+.1357	1.95	+.3023
.2	- .7279	.8	-.1009	1.4	+.1521	2.0	+.3135
.25	- .627	.85	-.0735	1.45	+.1681	2.05	+.3247
.3	- .5445	.9	-.0477	1.5	+.1834	2.10	+.3356
.35	- .4748	.95	-.0232	1.55	+.1982	2.15	+.3462
.4	- .4144	1.0	0	1.6	+.2126	2.20	+.3566
.45	- .3611	1.05	+.0221	1.65	+.2265	2.25	+.3668
.5	- .3135	1.10	+.0431	1.7	+.2400	2.30	+.3767

Table XII. Corrective Power Factor where

$$Z = -1.0414 \text{ Log} \left(\frac{e.h.p.}{E.H.P.}\right). \quad \text{Log I.H.P.}_y = \text{Log I.H.P.} + Z$$

For vessels of Class 8 submerged and of class 9, with surface and submerged, the values of Log A_v are given in Table XIV.

Knots V and v	3	4	5	6	7	8	9	10
Log A	1.94	2.17	2.36	2.53	2.66	2.79	2.91	3.01
Knots V and v	11	12	13	14	15	16	17	18
Log A	3.12	3.23	3.33	3.43	3.52	3.60	3.68	3.76
Knots V and v	19	20	21	22	23	24	25	26
Log A	3.83	3.89	3.95	4.0	4.05	4.1	4.14	4.18
Knots V and v	27	28	29	30	31	32	33	34
Log A	4.22	4.25	4.28	4.31	4.34	4.36	4.38	4.4
Knots V and v	35	36	37	38	39	40	41	42
Log A	4.43	4.44	4.46	4.47	4.48	4.49	4.50	4.51
Knots V and v	43	44	45	46	47	48	49	50
Log A	4.52	4.53	4.54	4.55	4.56	4.57	4.58	4.59
Knots V and v	51	52	53	54	55	56	57	58
Log A	4.60	4.61	4.62	4.63	4.64	4.65	4.66	4.67
Knots V and v	59	60	61	62	63	64	65	66
Log A	4.68	4.69	4.70	4.71	4.72	4.73	4.74	4.75

Table XIII.—Logs A_V for Vessels all Classes and Log A_v for Classes 1, 2, 3, 4, 5, 6, 7, 8 on the Surface and 10.

CAVITATION.

In discussing this subject, it must be divided into two conditions, as follows:

1. Suppose the propeller be designed for one particular condition of resistance of the vessel, then the performance curves of Power-Speed and Revolutions-speed will all pass through the design point without cavitation. As the speed increases, however, the resistance increases with it, but in a

Knots V	3	4	5	6	7	8	9	10
Log A_v	1.16	1.55	1.9	2.18	2.4	2.57	2.72	2.85
Knots V	11	12	13	14	15	16	17	18
Log A_v	2.96	3.07	3.17	3.26	3.35	3.44	3.52	3.6
Knots V	19	20	21	22	23	24	25	26
Log A_v	3.66	3.74	3.8	3.85	3.9	3.95	4.0	4.05
Knots V	27	28	29	30	31	32	33	34
Log A_v	4.08	4.09	4.15	4.18	4.21	4.24	4.26	4.28
Knots V	35	36	37	38	39	40	41	42
Log A_v	4.30	4.32	4.33	4.34	4.35	4.36	4.37	4.38
Knots V	43	44	45	46	47			
Log A_v	4.39	4.40	4.41	4.42	4.43			

Table XIV.—Log A_v for Vessels of Class 8 Submerged, and of Class 9 both Surface and Submerged.

greater degree, unless the vessel planes, until finally the flow of water to the propeller to supply the thrust required becomes greater than the combined atmospheric and hydraulic head on the water can produce. The result is that the propeller, instead of traveling smoothly along the suction column of the propeller, begins to cut across it and break the column apart,

consuming additional power and producing heavy vibrations. This phenomenon is designated as "*Cavitation*" or "*Cavitation of the Suction Column.*"

2. Suppose this same vessel be loaded down gradually so that with constant power the speed will gradually be reduced. This reduction in speed will be accomplished without any reduction in propulsive efficiency until a certain effective thrust be attained, after which any further reduction in speed will be accompanied by a reduction in propulsive efficiency. This phenomenon may be designated as "*Cavitation* or *Dispersal of the Thrust Column,*" and is accompanied by loss in efficiency and more or less vibration, depending upon the amount of reduction of speeds below those corresponding to the limiting thrusts.

Both of these conditions for "Cavitation" are shown by Fig. 4, where the curve of critical (limiting) thrusts are shown, marked C, Curve of Critical Thrusts, *E.T.*

The well authenticated limits of safety are between the curves A and C, although values of $v \div V$ above A may be used with probable safety, but to use such values in designing usually produces low values of $P \div D$ and is therefore to be avoided. The curve A gives thrusts 30 per cent lower than those of C, while Curve B corresponds to thrusts 15 per cent lower than C. Curve D at values of e.h.p. $\div$ E.H.P. up to .4 corresponds to conditions of maximum efficiency obtained by any vessels of which records are available, while the upper part of the curve from .9 upward give these same conditions for records at these high loads. This curve approximates closely to the curve corresponding to "Powers varying as the cubes of the speeds." Actual performance curves, however, approximate very closely to Curves A, D, C and the curves of $M = \left(\frac{e.t.}{E.T.}\right)^x$ shown, in their characteristics. The point corresponding to e.h.p $\div$ E.H.P. $=$ unity, and $v \div V =$ unity, is the point of Basic Condition of the propeller.

APPLICATION OF FIG. 4, TO CONDITION (1).

Suppose a propeller be of such proportions for the work it has to perform as to plot in its performance well above the curve of critical thrusts. As the speed increases, the value of e.t. ÷ E.T. will increase, but so long as there is no abnormal increase in e.t. ÷ E.T. as compared with the increase in v, the performance curve extended will still plot in the safety zone until the approximate value e.t. ÷ E.T. = 1.15 is reached. After passing this value, the combined hydraulic and atmospheric heads producing hydraulic flow to the propeller becomes insufficient and the propeller commences to cut across the water forming the suction supply column, cutting this column off in thin layers, so that there are a succession of these thin layers following each other through the length of the propeller. This discontinuity of the water supply produces the phenomenon commonly spoken of as "Cavitation." It produces a loss in power, gradually increasing in amount as the value of e.t. ÷ E.T. increases above 1.15, and vibration occurs increasing as the value e.t. ÷ E.T. increases.

Cavitation will have the following effects upon estimate of power where effective power is given, upon estimate of effective power where indicated or shaft horsepower is given, and upon the equation for apparent slip.

(a) *e.h.p. Given to Find I.H.P.$_p$ or S.H.P.$_d$.*

Suppose e.h.p. ÷ E.H.P. = 1.02, and v ÷ V = .8, K = 1.1. The value of M for these values of e.h.p. ÷ E.H.P. and v ÷ V is 1.111.

Z for e.h.p. ÷ E.H.P. = +.011 (Table XII).

Log I.H.P.$_p$ = Log I.H.P. + Z,

Log I.H.P.$_d$ (without cavitation) = Log I.H.P.$_p$ × Log K,

Log I.H.P.$_d$ (with cavitation) =

Log I.H.P.$_p$ + Log K + Log M.

(b) *I.H.P.$_d$ Given to Find e.h.p.*, v ÷ V = known value, e.h.p. ÷ E.H.P. unknown. Calling the corrective Power Factor for that value of e.h.p. ÷ E.H.P. at the intersection

of the value $v \div V$ with the Critical Thrust Curve Z_1, and Z the value for the actual value of the e.h.p. ÷ E.H.P. delivered;—

$Z = Z_1 - \text{Log M}$, and M will have the value that satisfies this equation and the equation for Log I.H.P.$_a$ as given below.

(c) *To Find Revolutions*, either e.h.p. or I.H.P.$_a$ being known, then

$$s = S\frac{\text{I.H.P.}_a \times A_v}{\text{I.H.P.} \times A_v} = S\frac{M \times K \times A_v}{10^2 \times A_v},$$

where Z_1 = the value of Z corresponding to that value of e.h.p. ÷ E.H.P. where $v \div V$ crosses the curve of Critical Thrusts, and where Z for the actual value of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ being delivered is expressed by $Z = Z_1 + \text{Log M}$.

APPLICATION OF FIG. 4 TO CONDITION 2.

Water in passing through a propeller has its direction of flow gradually changed from the direction with which it originally enters the propeller at the leading edges of the propeller blades, until on leaving the propeller it passes off in the direction of the generatrices of an infinite number of right hyperboloids of revolution, the outer limiting one having for its base the propeller disc. The efficiency of propulsion realized, will, among other causes, depend upon the distance of the minimum diameter of this limiting hyperboloid abaft the propeller and upon the area of this minimum cross section; the more nearly the limiting hyperboloid approaches to a cylinder, the greater will be the efficiency of propulsion. As the resistance of a ship is increased by excessive loading, foul bottom or head winds and seas, the slower will be the speed realized for any given power and the closer to the propeller and the smaller in area will become the minimum belt area of the thrust hyperboloid. The efficiency of propulsion will remain practically constant throughout quite a large range of speeds until the speed becomes such that the thrusts for any given speed and effective horsepower plots the propeller upon

the Curve of Critical Thrusts, *E.T.* Any further decrease in speed will be accompanied by a decrease in efficiency exactly as in *Condition* 1, and the same corrective formulas as given in the preceding section will apply.

GENERAL FORMULAS FOR CORRECTION OF POWER FOR "CAVITATION" AND "DISPERSAL OF THE THRUST COLUMN."

$\frac{\text{e.h.p.}}{\text{E.H.P.}}$ = Actual Load Factor.

$\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$ = Load Factor corresponding to point on Critical Thrust Curve where $v \div V$ crosses that curve.

Z = Power Exponent corresponding to $\frac{\text{e.h.p.}}{\text{E.H.P.}}$.

Z_1 = Power Exponent corresponding to $\frac{\text{e.h.p.}_1}{\text{E.H.P.}}$.

Z and Z_1 regarded as positive.

(1) $\frac{\text{e.h.p.}}{\text{E.H.P.}} > 1$, $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} > 1$.

$\text{Log } M = Z - Z_1$.

$\text{Log I.H.P.}_a = \text{Log I.H.P.} + \text{Log } K + 2Z - Z_1$.

(2) $\frac{\text{e.h.p.}}{\text{E.H.P.}} > 1$, $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} = 1$.

$\text{Log } M = Z$.

$\text{Log I.H.P.}_a = \text{Log I.H.P.} + \text{Log } K + 2Z$.

(3) $\frac{\text{e.h.p.}}{\text{E.H.P.}} > 1$, $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} < 1$.

$\text{Log } M = Z + Z_1$.

$\text{Log I.H.P.}_a = \text{Log I.H.P.} + \text{Log } K + 2Z + Z_1$.

(4) $\frac{\text{e.h.p.}}{\text{E.H.P.}} = 1$, $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} < 1$.

$\text{Log } M = Z_1$.

$\text{Log I.H.P.}_a = \text{Log I.H.P.} + \text{Log } K + Z_1$.

(5) $\frac{\text{e.h.p.}}{\text{E.H.P.}} < 1$, $\frac{\text{e.h.p.}_1}{\text{E.H.P.}} < 1$.

$\text{Log } M = Z_1 - Z.$

$\text{Log I.H.P.}_d = \text{Log I.H.P.} + \text{Log } K + Z_1 - 2Z.$

ANALYSIS AND DESIGN OF PROPELLERS.

By use of the following formulas it is possible, having given the characteristics of a propeller, including form of projected area, and the general characteristics of the hull behind which the propeller is to work, including the curve of e.h.p. — Speed, to reduce the propeller to standard conditions and to make a very close estimate of the power and revolutions for any given speed under the conditions of resistance represented by the curve of e.h.p.

Conversely, having the general characteristics of a hull and the curve of e.h.p. — Speed, either from model tank trials or estimated, it is possible to design the propeller and estimate the power and revolutions for any required speeds, or having a fixed power and revolutions, to estimate the speed which will be obtained.

FORMS FOR ANALYSIS OF PROPELLER.

Form 1. Curve of effective (tow-rope) horsepower available.

Propeller blades of Standard Projected Area Ratio Form.

(1) D = Diameter of Propeller, in feet.

(2) P = Pitch of Propeller, in feet.

(3) *P.A.* ÷ *D.A.* = Total Projected Area Ratio of Propeller outside of .2 D circle.

(4) P.A. ÷ D.A. = Basic Projected Area Ratio = ¾ (*P.A.* ÷ *D.A.*) for 4-bladed propeller = 3/2 (*P.A.* ÷ *D.A.*) for 2-bladed propeller.

(5) T.S. = Basic Tip Speed of Propeller, in feet, per minute for P.A. ÷ D.A. (Table VIII).

(6) P × R = (T.S. × P.) ÷ πD.

(7) S.B.C. = Slip Block coefficient of hull of vessel to which propeller is applied; derived from equation for type of hull and relative position of propeller.

(8) $1 - S = 1 -$ Apparent Basic Slip, taken from Table VIII, for P.A. ÷ D.A. and S.B.C.

(9) $V = \{ P \times R \times (1 - S) \} \div 101.33 =$ Basic Speed, in knots per hour.

(10) $I.T._D.$ = Basic Indicated Thrust, in pounds per square inch of disc area, including the hub. Taken from Table VIII, for P.A. ÷ D.A.

(11) $I.H.P. = (D^2 \times I.T._D \times P \times R) \div C =$ Basic Indicated Horsepower,

where $C = \begin{cases} 389 & \text{— for 2-bladed propellers.} \\ 291.8 & \text{— for 3-bladed propellers.} \\ 252.4 & \text{— for 4-bladed propellers.} \end{cases}$

(12) P.C. = Basic Propulsive Coefficient for *P.A. ÷ D.A.*

(13) E.H.P. = I.H.P. × P.C. = Basic Effective (Tow-rope) Horsepower delivered by the propeller.

(14) S.H.P. = I.H.P. × .92 = Basic Shaft Horsepower delivered to the propeller.

TO FIND POWER FOR ANY DESIRED EFFECTIVE (TOW-ROPE) POWER.

(15) v = Any desired speed of ship.

(16) e.h.p. = Total net effective (two-rope) horsepower to be delivered by one propeller for speed v.

(17) e.h.p. ÷ E.H.P. = Load factor.

(18) $z = -1.0414 \log \left(\frac{\text{e.h.p.}}{\text{E.H.P.}}\right) = 1.0414 \log \left(\frac{\text{E.H.P.}}{\text{e.h.p.}}\right)$ = Power corrective factor for e.h.p. ÷ E.H.P. (Table XII).

(19) $I.H.P._p$ or $S.H.P._p$ = (I.H.P. or S.H.P.) ÷ 10^z = Indicated or Shaft horsepower required to deliver e.h.p. without *Thrust Deduction*.

(20) K = Thrust Deduction factor for S.B.C. as explained under section on Thrust Deduction and relative position of propeller.

(21) I.H.P.$_d$ or S.H.P.$_d$ = K $\times$ (I.H.P.$_p$ or S.H.P.$_p$) = Gross Indicated or Shaft horsepower required to deliver e.h.p.

(22) *p.c.* = e.h.p. $\div$ I.H.P.$_d$ = Final propulsive coefficient.

TO FIND REVOLUTIONS.

(23) Log A_V = Speed Factor for V (Table XIII).

(24) Log A_v = Speed Factor for v (Table XIII).

(25) $s = S\,\frac{\text{I.H.P.}_d \times A_V}{\text{I.H.P.}_c \times A_v} = S\,\frac{\text{S.H.P.}_d \times A_V}{\text{S.H.P.}_c \times A_v}$ = when no cavitation exists,

$\therefore S\,\frac{K \times A_v}{10^2 \times A_V}$ = Apparent slip for speed v.

(26) $R_d = \frac{101.33 \times v}{P \times (1 - s)}$ = Estimated Revolutions for Power $\left\{\begin{matrix}\text{I.H.P.}_d \\ \text{S.H.P.}_d\end{matrix}\right\}$ and speed v.

VARIATIONS FROM ABOVE FORM.

(a) Engine to be designed for a given power to find estimated speed and revolutions at that speed.

(15) I.H.P.$_d$ or S.H.P.$_d$ = Gross designed Power of Engine (or on one shaft).

(16) K = *Thrust Deduction* factor for S.B.C. and relative position of propeller.

(17) I.H.P.$_p$ or S.H.P.$_p$ = (I.H.P.$_d$ or S.H.P.$_d$) $\div$ K = Net Power of Engine (or on one shaft) actually producing propulsive thrust.

(18) Z = (Log I.H.P. or Log S.H.P.) $-$ (Log I.H.P.$_p$ or Log S.H.P.$_p$) = Power corrective factor.

(19) (e.h.p. $\div$ E.H.P.) = $\frac{1}{10^z \div 1.0414}$ = Load Factor.

(20) e.h.p. = E.H.P. × (e.h.p. ÷ E.H.P.) = Net effective (tow-rope) horsepower delivered by one propeller.

(21) N = Number of Propellers.

(22) N × e.h.p. = Total effective horsepower delivered.

(23) v = Speed corresponding to N × e.h.p. from curve of effective tow-rope horsepower.

Using I.H.P.$_d$ or S. H.P.$_d$ and I.H.P. or S.H.P. and Logs A_v, log A_v (the latter for the v of (23)), find s and R_d as before.

(b) Engine to be designed for a given power. Speed estimated at a given value, but no effective horsepower for this speed given. To find effective horsepower which will be delivered at this speed and the corresponding revolutions.

(15) I.H.P.$_d$ or S.H.P.$_d$ = Gross designed Power of Engine (or on shaft).

(16) K = Thrust Deduction factor for S.B.C. and relative position of propeller.

(17) I.H.P.$_p$ or S.H.P.$_p$ = (I.H.P.$_d$ or S.H.P.$_d$) ÷ K = Net Power of Engine (or on shaft) actually producing propulsive thrust.

(18) Z = (Log I.H.P. or Log S.H.P. — (Log I.H.P.$_p$ or Log S.H.P.$_p$) = Power corrective factor.

(19) e.h.p. ÷ E.H.P. = Load factor corresponding to Z.

(20) e.h.p. = E.H.P. × (e.h.p. ÷ E.H.P.) = Net Effective (tow-rope) horsepower delivered by one propeller.

(21) v = Given designed speed.

From the data of I.H.P.$_d$ or S.H.P.$_d$ and v, find s and R_d as before.

It is evident that the values of s and R_d will depend entirely upon the correctness of the estimates of power, slip block coefficient and speed. Should the power estimate be too low for the speed, the revolutions and speed obtained with the designed power of the engine will be lower than the designed revolutions and the estimated speed of the ship, and *vice versa*. Should the slip block coefficient used be too heavy, the estimated revolutions will be too low.

The above forms are those to be used where blades are of standard Form of Projected Area Ratio and where no cavitation of the suction column to nor dispersal of the thrust column from the propeller exists.

It will now be necessary to give the characteristics of the Standard Forms referred to.

STANDARD FORMS OF PROJECTED AREA RATIOS.

The projected area forms of all propellers, no matter of what contour, may be swung around in circular arcs until the areas are symmetrically arranged on each side of a center line drawn perpendicular to the axis of the propeller shaft, and the original area of the projection and its radial distribution of surface will be retained. If a form of blade of constant percentage radial distribution of projected surface is adopted, no matter what percentage of the disc area represents the projected area, the relative distribution of projected surface will be a constant and therefore one of the most frequent variables met with in propeller design, a variable which seriously affects efficiency and revolutions, will be eliminated. To eliminate another variable to as great an extent as possible, the blade sections on the circular arcs through the projections have also been made of standard form as nearly as possible.

STANDARD PROJECTED AREA FORMS.

The Standard Projected Area Ratio Form adopted is that used for a large number of propellers in the United States Navy, whose records of performances averaged very high and, in general, exceeded those given by any other forms.

These characteristics are set forth in the following table, where with a propeller radius of unity, the chords of the arcs between the middle line of the projected area and either leading or following edge of the projection at each of the values .2, .3, .4, .5, etc., of the radius are given for each value of Projected Area Ratio of a 3-bladed propeller, from P.A. ÷ D.A. = .15 to P.A. ÷ D.A. = .7.

Table XV. Table of Cords of Area between Middle Radial Line of Projection of Blade and Bounding Edge.

P. / D A.	Decimal Parts of Radius with Cords Below										
	.2	.3	.4	.5	.6	.7	.8	.9	.925	.95	.975	.9875
.15	.042	.064	.087	.107	.120	.128	.126	.106	.084	.08	.059	.043
.20	.055	.084	.114	.140	.159	.171	.168	.139	.104	.103	.078	.059
.25	.069	.107	.142	.173	.198	.214	.206	.174	.157	.133	.097	.069
.30	.081	.128	.170	.207	.236	.253	.250	.210	.190	.161	.120	.088
.35	.097	.150	.199	.243	.274	.296	.293	.246	.223	.192	.143	.105
.40	.106	.170	.227	.277	.316	.338	.333	.278	.251	.214	.166	.118
.45	.123	.191	.253	.308	.353	.378	.374	.313	.284	.243	.178	.127
.50	.136	.210	.279	.341	.390	.417	.416	.346	.313	.269	.195	.143
.55	.148	.230	.306	.372	.425	.456	.456	.383	.348	.293	.216	.155
.60	.159	.249	.333	.405	.463	.499	.497	.413	.377	.322	.238	.184
.65	.172	.270	.358	.437	.497	.530	.537	.445	.403	.343	.255	.200
.70	.184	.288	.382	.466	.533	.578	.576	.481	.437	.375	.280	.220

For a 4-bladed propeller the standard form corresponding to the total projected area ratio will be that for three-fourths of that ratio, while for a 2-bladed propeller it would be that corresponding to three-halfs of the total projected area ratio.

In laying down the projected area form for any desired projected area ratio it must be borne in mind that all projected area ratios are outside of the .2 R circle, and any variation from this in the actual blade due to variation of hub diameter may be neglected, as may also any slight widening of the blade inside of the .7 R circle which may be made in order to reduce the thickness of the blade near the hub by increasing the width.

The lengths of the chords of the half arcs for any diameter propeller, D = diameter, will equal length of chord as given in the Table for each fraction of the radius and the particular projected area ratio desired multiplied by the radius of the propeller.

Thus, suppose the diameter of the propeller to be 10 feet and the desired projected area ratio to be .6, the length of the chord of the half arc at .7R will be $.499 \times 5 = 2.495$.

The greatest circular width of the projected area form of the Standard Form is at the .7R radius.

STANDARD FORM OF BLADE SECTIONS.

The Standard blade-section used is that having the driving face a true helical surface generated by the generatrix of the pitch surface of the propeller, while the back of the propeller is formed by a circular arc, the edges of the blades being made as sharp as possible yet being thickened up sufficiently to give durability.

The approximate ratio of thickness to width of developed circular arc at .2R from the center of the hub for blades of manganese bronze of 60,000 pounds tensile strength should be approximately according to the following Table:

P.A.÷D.A. (3Blades)	.15	.2	.25	.3	.35	.4
Thickness÷Width	.19	.2	.197	.193	.187	.18

P.A.÷D.A. (3Blades)	.45	.5	.55	.6	.65
Thickness÷Width	.171	.16	.15	.14	.127

Table XVI. Table of Blade Thicknesses.

P÷D	.8	1.0	1.2	1.4	1.6	1.8	2.0
P.A÷ D.A.	Values of E.A.÷D.A.						
.1	.11	.117	.125	.132	.140	.148	.156
.15	.165	.175	.186	.197	.209	.222	.235
.2	.221	.235	.250	.266	.282	.298	.315
.25	.277	.296	.315	.334	.353	.372	.390
.3	.334	.356	.378	.400	.422	.444	.465
.35	.390	.415	.440	.465	.490	.515	.540
.4	.445	.473	.500	.528	.555	.583	.610
.45	.497	.527	.557	.587	.617	.647	.678
.5	.550	.583	.615	.648	.680	.713	.745
.55	.600	.634	.669	.703	.738	.772	.807
.6	.650	.686	.723	.758	.794	.830	.866
.65	.700	.738	.775	.813	.850	.888	.927
.7	.75	.79	.83	.87	.91	.95	.99

*3 Blades

Table XVII. Projected Area and Corresponding Developed Area Ratios for Different Values of Pitch Divided by Diameter.

VARIATIONS OF PROJECTED AREA FORMS FROM STANDARD.

Blade forms of projected area may vary in three different manners from the Standard Forms, as follows:

1. Projected area form widening gradually as the radius increases. Circumferential bounding edge circular with corners rounded. Will be known as *Fan-Shaped Blades.*

2. Projected area form having greatest half arc chord at greater than .7R from the center of the hub. Will be known as *Broad-Tipped Blades.*

3. Projected area form having greatest half arc chord at less than .7R from the center of the hub. Will be known as *Narrow-Tipped Blades.*

The three types are shown in Figs. 1, 2 and 3. In each case the actual propeller can be referred to a Basic propeller having, in Case 1, a projected area outside the .2R equal to the projected area of the actual propeller outside of this same limiting circle. The Basic propeller of diameter D_1 must be

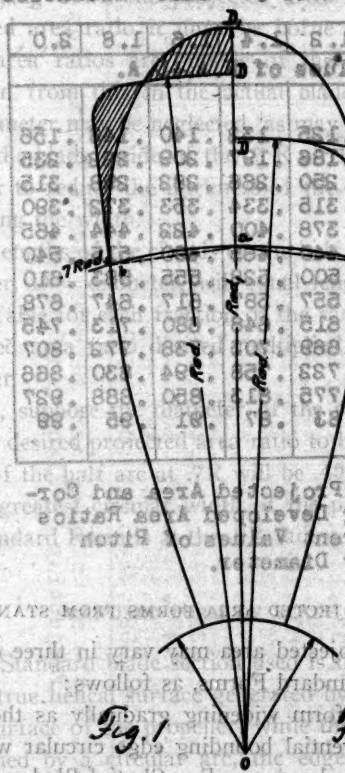


Fig. 1 Fig. 2

found by assuming various diameters and laying down the projected area forms for those values of projected area ratios which will coincide closely with the actual rectified form of projected area for the greater part of the blade and whose area outside the tip of the actual blade, shown hatched, will

equal the hatched area of the actual blade. In Cases 2 and 3, the greatest length of chord of the half arcs through projected area should be ascertained. The actual radius from

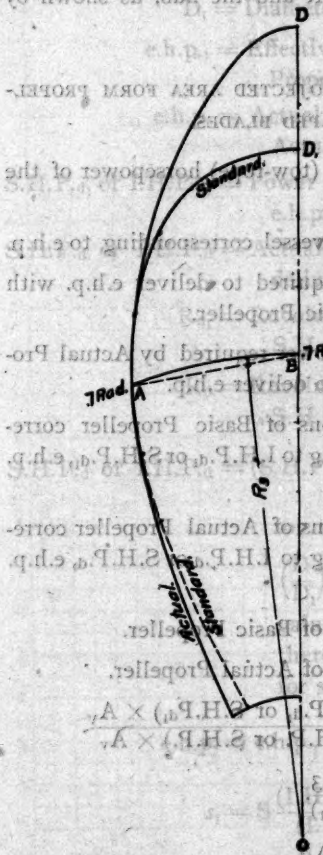


Fig. 3.

the center of the hub to this arc, multiplied by two and divided by .7, will give the diameter of the Basic propeller. Dividing the length of the chord by the radius of the Basic propeller

will give the standard length of chord at .7R for unity radius, and the corresponding value of P.A. ÷ D.A., for the Basic 3-bladed propeller can then be picked out or interpellated from Table XV. Slight departures of the standard form from the actual form between the .7R and the hub, as shown by Fig. 3, may be neglected.

ANALYSIS OF NON-STANDARD PROJECTED AREA FORM PROPELLERS, FAN-SHAPED BLADES.

Let e.h.p. = Effective (tow-rope) horsepower of the vessel.

v = Speed of vessel corresponding to e.h.p.

$S.H.P._{d_1}$ or $I.H.P._{d_1}$ = Power required to deliver e.h.p. with the Basic Propeller.

$S.H.P._d$ or $I.H.P._d$ = Actual power required by Actual Propeller to deliver e.h.p.

R_{d_1} = Revolutions of Basic Propeller corresponding to $I.H.P._{d_1}$ or $S.H.P._{d_1}$, e.h.p. and v.

R_d = Revolutions of Actual Propeller corresponding to $I.H.P._d$ or $S.H.P._d$, e.h.p. and v.

D_1 = Diameter of Basic Propeller.

D = Diameter of Actual Propeller.

Then
$$s_1 = S \times \frac{(I.H.P._{d_1} \text{ or } S.H.P_{d_1}) \times A_V}{(I.H.P. \text{ or } S.H.P.) \times A_v}.$$

$$R_{d_1} = \frac{v \times 101.33}{P \times (1 - s_1)}.$$

$$R_d = R_{d_1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{2}}.$$

$$(I.H.P._d \text{ or } S.H.P._d) = (I.H.P._{d_1} \text{ or } S.H.P._{d_1}) \times \frac{D_1}{D}.$$

and the propulsive coefficients will vary as the diameters.

BROAD-TIPPED AND NARROW-TIPPED BLADES

Let the diameter of the circular arc of greatest projected area length $= D_0$.

D = Diameter of Actual Propeller.

D_1 = Diameter of Basic Propeller $= D_0 \div .7$.

$e.h.p._1$ = Effective horsepower delivered by Basic Propeller.

$e.h.p.$ = Actual effective to be delivered by Actual Propeller.

$S.H.P._{d_1}$ or $I.H.P._{d_1}$ = Power of Basic Propeller to deliver $e.h.p._1$, at speed v.

$S.H.P._d$ or $I.H.P._d$ = Actual Power of Actual Propeller to deliver e.h.p. at speed v.

R_{d_1} = Revolutions of Basic Propeller with $S.H.P._{d_1}$ or $I.H.P._{d_1}$ at speed v.

R_d = Revolutions of Actual Propeller with $S.H.P_d$ or $I.H.P._d$ and v.

$S.H.P._d$ or $I.H.P._d = (S.H.P._{d_1}$ or $IHP_{d_1}) \times \left(\frac{D}{D_1}\right)^2$. This factor varies apparently from $\left(\frac{D}{D_1}\right)^2$ to $\left(\frac{D}{D_1}\right)^{3/2}$, the latter giving the greater values of $I.H.P._d$ or $S.H.P_d$, and therefore giving a greater lee-way for safety in the estimate.

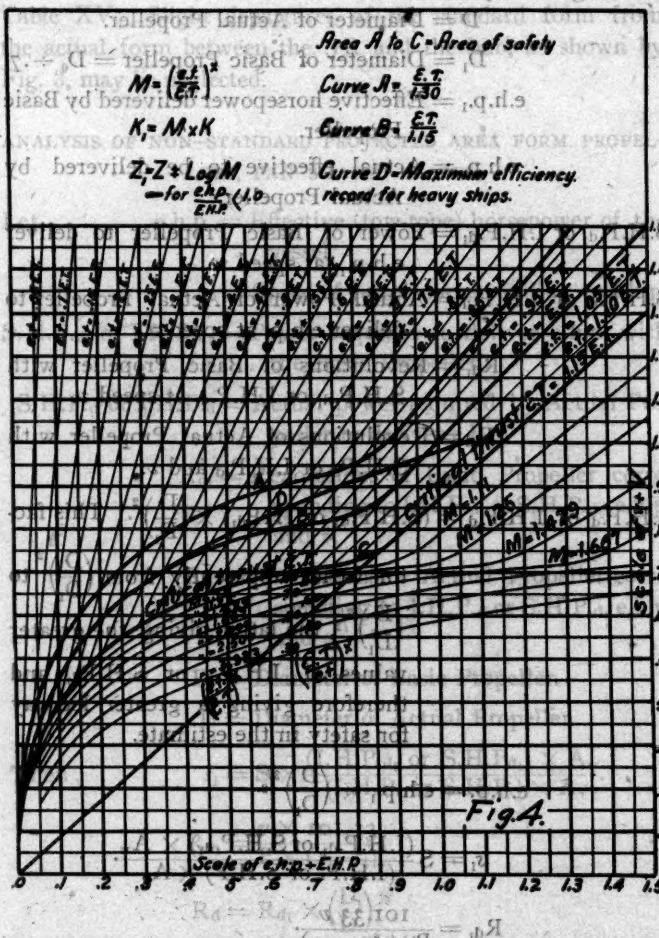
$$e.h.p._1 = e.h.p._1 \times \left(\frac{D}{D_1}\right)^{3/2}.$$

$$s_1 = S\frac{(I.H.P._{d_1} \text{ or } S.H.P._{d_1}) \times A_T}{(I.H.P. \text{ or } S.H.P) \times A_{T_1}}.$$

$$R_{d_1} = \frac{101.33\ v}{P \times (1 - s_1)}.$$

$$R_d = R_{d_1} \times \left(\frac{D_1}{D}\right)^{1/4}.$$

Should the Basic Results plot in the cavitating area of Fig. 4, the corrections for Power or for effective power de-


Area A to C = Area of safety
$M = \left(\frac{n\sqrt{j}}{E.T.}\right)^2$
Curve A = $\frac{E.T.}{1.30}$
$K_1 = M \times K$
Curve B = $\frac{E.T.}{1.15}$
$Z_1 = Z + \text{Log } M$
Curve D = Maximum efficiency record for heavy ships.
—for $\frac{e.h.p.}{E.H.P.}$ (1.0
Scale of e.h.p. ÷ E.H.P.
0 .1 .2 .3 .4 .5 .6 .7 .8 .9 1.0 1.1 1.2 1.3 1.4 1.5
Fig. 4.

livered must be applied to the Basic Propeller estimates and the values for the actual propeller then be obtained from these corrected Basic values.

FACTS CONCERNING THE PRODUCTION OF CAVITATION.

(1) To lighten the load on a vessel with a given propeller delays cavitation and *vice versa.*

(2) To increase the projected area ratio of a propeller, pitch and diameter remaining constant, produces a delay in cavitation and in dispersal of the thrust column and *vice versa.*

(3) To increase the pitch of a given propeller tends to expedite cavitation and *vice versa.*

(4) To decrease the diameter of a given propeller, the pitch remaining constant, tends to produce earlier cavitation and dispersal of the thrust column.

(5) The existence of thrust deduction, in addition to increasing the power necessary for propulsion, decreases the speed and net effective thrust at which cavitation will occur.

(6) Where the hull of a vessel is of such underwater form as to produce a heavy wake, the speeds at which cavitation and dispersal of the thrust column occurs will be higher than if no wake existed, on account of the "wake gain."

(7) Where insufficient tip clearance exists between the propeller and the hull, increases in effective horsepower and speed of vessel are accompanied by a gradual increase in the thrust deduction, which latter increase produces earlier cavitation.

(8) Where sections across propeller blades are of such form as to give a free flow of water past the sections with no tendency to form eddies, but where the sections are abnormally thick as compared with their width, the abnormal thickness produces an actual pitch considerably greater than the normal pitch and cavitation is expedited.

(9) Where blade sections are very thick as compared with their width and where the entering and leaving edges are of such form as to produce eddying of the water flowing past the sections at the higher speeds, the thrust per revolution will be increased due to the higher actual pitch produced by the thickness of the sections, and the power required per revolu-

tion will increase in greater proportion than the effective thrusts. As the blade speeds increase eddies begin to form, and this formation of eddies is accompanied by a still further exaggeration of power and all of the phenomena of cavitation, and this will occur at lower thrusts and speeds than would be the case were the sections normally fine.

DESIGN OF THE PROPELLER.

Limitations of Load Factors imposed by the S.B.C. of the vessel and by the speed, as applied to all conditions of design are given in Table XVIII.

SLIP BLOCK COEFFICIENT													
.3		.4		.5		.6		.7		.8		.9	
d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.	d	c.h.p. ÷ E.H.P.
12	.015	6	.009	0	–	0	–	0	–	0	–	0	–
14	.022	8	.015	2	.006	2	.007	1	.003	1	.005	1	.005
16	.034	10	.022	4	.01	4	.01	2	.007	2	.011	2	.011
18	.053	12	.038	6	.013	6	.025	3	.01	3	.02	3	.02
20	.083	14	.063	8	.022	8	.055	4	.018	4	.03	4	.044
22	.12	16	.10	10	.045	10	.11	5	.03	5	.05	5	.085
24	.175	18	.155	12	.09	12	.205	6	.045	6	.085	6	.14
26	.24	20	.233	14	.163	14	.345	7	.07	7	.135	7	.213
28	.34	22	.335	16	.26	16	.53	8	.105	8	.19	8	.325
30	.5	24	.485	18	.395	18	.765	9	.15	9	.27	9	.48
32	.69	26	.685	20	.585	20	1.05	10	.217	10	.39	10	.69
34	.9	28	.91	22	.815	22	1.36	11	.295	11	.53	11	.9
36	1.11	30	1.18	24	1.07	24	1.7	12	.394	12	.69	12	1.15
38	1.35	32	1.41	26	1.34			13	.505	13	.885	13	1.43
40	1.6	34	1.7	28	1.63			14	.635	14	1.06	14	1.7

Table XVIII. Limiting Maximum Values of c.h.p.÷E.H.P. for Various Values of S.B.C. and of v.

These limit values are only approximate and may be exceeded by about twenty-five per cent. The speed used should be the speed expected under conditions of maximum hull resistance. The propeller obtained will be the approximate smallest and most efficient propeller suited to meet these conditions.

Limiting values of v ÷ V.—These for any load factor c.h.p. ÷ E.H.P., should never be less than those correspond-

ing to the *Critical Thrusts, E.T.* (Fig. 4) for this load factor. In cases where it may be absolutely necessary to use lower values of $v \div V$, the augment of thrust deduction (M) must be brought into the computations as explained under the general subject of "Cavitation."

PROBLEMS IN PROPELLER DESIGN.

These may be classed under two general heads, namely:

(1) Problems of sufficient data.
(2) Problems of insufficient data.

Under the first head are grouped all problems in which full hull data and curve of e.h.p. — Speed are given. Designed power of engines and revolutions desired at this power may also be supplied or it may be desired to obtain an estimate of power with the best revolutions for any given speed. Such problems are those which a shipbuilder should recommend to his customer if he has the best interests of the customer at heart.

Under the second head are grouped all problems in which hull data, more or less full, is generally supplied, but in which no curve of e.h.p. — speed is available. The designed power of the engines and their revolutions at this power are given together with the speed which it is expected to obtain with this power. Problems under this head are very unsatisfactory, as should the estimate of speed be wrong the revolutions actually obtained may vary widely from those desired, increasing should the speed be greater than the estimated, and decreasing should it be less, while should the actual speed be much less than the estimate, the danger zone below the curve of *Critical Thrusts*, Fig. 4, may be entered and a corresponding loss in propulsive efficiency be encountered.

FORMS FOR DESIGN COMPUTATIONS (STANDARD BLADES).

(1) Speed, effective horsepower, limiting maximum diameter given, to find Projected Area Ratio, indicated or shaft

horsepower, diameter, pitch, revolutions and propulsive efficiency; percentage variation of loading of vessel small. This condition applies to Naval and Passenger Vessels.

(1) D Max. (2); (3) (4) decreasing values.

(2) e.h.p. ÷ E.H.P. Max. allowed for S.B.C. and v (Table XVIII).

(3) e.h.p. Constant.

(4) E.H.P. = e.h.p. ÷ $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ Constant.

(5) v (knots) Constant designed speed.

(6) v ÷ V Constant for value of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ on Curve D, Fig. 4.

(7) $V = v \div \frac{v}{V}$ Constant.

(8) $E.T._p \times \frac{P.A.}{D.A.} = \frac{C \times E.H.P.}{D^2 \times V}$.

$$C = \begin{cases} 2.491 \text{ for 4-bladed screws.} \\ 2.88 \text{ for 3-bladed screws.} \\ 3.84 \text{ for 2-bladed screws.} \end{cases}$$

(9) P.A. ÷ D.A. (Basic and 3 blades, Total Proj. Area Ratio).

(10) $\begin{cases} \frac{4}{3} \text{ P.A. ÷ D.A. (4 blades, Total Proj. Area Ratio).} \\ \frac{2}{3} \text{ P.A. ÷ D.A. (2 blades, Total Proj. Area Ratio).} \end{cases}$

(11) P.C. for $\left\{ \begin{matrix} \text{(9) 3 blades} \\ \text{(10) 4 and 2 blades} \end{matrix} \right\}$ (Table VIII).

(12) I.H.P. = E.H.P. ÷ P.C.

(13) S.H.P. = I.H.P. × .92.

(14) T.S. for (9) (Table VIII.)

(15) S.B.C. from equation for S.B.C. = Constant.

(16) 1 — S for (9) and (15) (Table VIII).

(17) $P = (\pi D \times V \times 101.33) \div \{T.S. \times (1 - S)\}$.

ESTIMATE OF POWER.

(18) Z for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ Table XII.

(19) $\text{I.H.P.}_p = \text{I.H.P.} \div 10^2$.

(20) K for S.B.C.

(21) $\text{I.H.P.}_d = \text{I.H.P.}_p \times K$.

When $v \div V$ for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ falls below or to the left of the Curve of Critical Thrusts (Fig. 4):

(22) M for $\frac{v}{V}$ and $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ (Fig. 4) (Log $M = Z_1 - Z$ where Z_1 is value for $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ where $\frac{v}{V}$ crosses *E.T.*).

(23) $\text{I.H.P.}_d = \text{I.H.P.}_p \times K \times M$.

ESTIMATE OF REVOLUTIONS.

(24) Log A_v for V (Table XIII).

(25) Log A_v for v (Table XIII or XIV).

(26) $s = S \frac{\text{I.H.P.}_d \times A_v}{\text{I.H.P.} \times A_v}$.

(27) $R_d = \frac{v \times 101.33}{P \times (1 - s)}$.

Should the revolutions obtained with the maximum diameter be higher than those considered desirable for the type of propelling engines to be fitted, the calculations should be continued, using the maximum diameter with decreasing values of $v \div V$ and e.h.p. $\div$ E.H.P., but retaining the value of *e.t.* $\div$ E.T. corresponding to the plotted point of the first cal-

calculation with maximum diameter plotted on Fig 4, at e.h.p. ÷ E.H.P. = 3, $v \div V = .67$. The value of *e.t.* ÷ E.T. as shown by the diagonal line through this point is .45. In the new calculations this value is kept constant, and points culation as taken from Fig. 4. Thus, suppose the original along the diagonal line are used Should the curve of Critical Thrusts be reached and the revolutions still be too high, then follow along this curve with decreasing values of e.h.p. ÷ E.H.P. for further calculations, but the change in revolutions along this curve will be small. Should the propeller finally obtained fall on the curve of Critical Thrusts, it will be in a precarious position, as any increase in resistance for a given speed will produce a corresponding decrease in efficiency of propulsion.

(2) Indicated or Shaft Horsepower of Engines and designed revolutions given, also (a) effective horsepower curve (sufficient data), or (b) estimated speed (insufficient data), together with hull characteristics, location of propellers and maximum diameter of propeller that can be carried.

Let it be supposed in this case that the revolutions are low and the power comparatively high, which will necessitate the use of the maximum diameter of propeller possible. Further, let it be supposed that the vessel is one subject to heavy variations in load, as in a collier or freight steamer. Call the speed estimated for the light condition v_1 and that for the loaded condition v_2. Should the propeller be designed for v_1 the revolutions would be seriously lowered when speed is reduced by loading to v_2, also, in the case of reduction gear ships, the tooth pressures in the gears would be unduly increased. Should the propeller be designed for v_2, the revolutions will unduly increase in the light condition and the engines or turbines be dangerously overspeeded. It would appear to be preferable, therefore, to design the propeller for safety at the v_2 condition, and for designed revolutions at a speed

$$v = \frac{v_1 + v_2}{2}$$

(2) FORM FOR CALCULATION FOR PROJECTIVE AREA RATIO, PROPULSIVE COEFFICIENT AND EFFECTIVE HORSEPOWER.

(1) D = Max. Diameter of Propeller that can be carried.

(2) v_2 = Est. Min. Speed under maximum load condition, with designed power.

(3) $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ = Safe load at v_2 for S.B.C. (Table XVIII).

(4) I.H.P.$_d$ = (S.H.P.$_d$ ÷ .92) = Designed Power.

(5) K = Thrust Deduction Factor.

(6) I.H.P.$_p$ = I.H.P.$_d$ ÷ K = Propulsive Horsepower.

(7) Z = Power Factor for e.h.p. ÷ E.H.P. (Table XII).

(8) I.H.P. = I.H.P.$_p$ × 10^z = Basic Power.

(9) $v_2 \div V$ = That value corresponding to $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ on Curve of Critical Thrusts (Fig. 4).

(10) $V = v_2 \div (v_2 \div V)$ = Basic Speed.

(11) $\text{I.T.}_d \div (1-S) = \frac{C \times \text{I.H.P.}}{D^2 \times V}$.

Where C = 3.84 for 2-bladed screws.
2.88 for 3-bladed screws.
2.491 for 4-bladed screws.

(12) P.A. ÷ D.A. = Proj. Area Ratio for 3 blades and for Basic (Table IX).

(13) ⅔ P.A. ÷ D.A. = Proj. Area Ratio for 2 blades (Table IX).

(14) ⅘ P.A. ÷ D.A. = Proj. Area Ratio for 4 blades (Table IX).

(15) P.C. = Basic Propulsive Cœfficient, taken from Table VIII for (12), (13) or (14), depending upon number of blades.

(16) E.H.P. = I.H.P. × P.C. = Basic Effective Horsepower.

(17) e.h.p. $= \text{E.H.P.} \times \frac{\text{e.h.p.}}{\text{E.H.P.}} =$ Effective horsepower delivered by I.H.P.$_a$ or S.H.P.$_a$.

(18) $v_1 =$ Speed for e.h.p. from effective horsepower curve, or estimated, for light condition.

(19) $v_2 =$ Speed for e.h.p. from effective horsepower curve, or estimated, for heavy condition.

(20) $v = \frac{v_1 + v_2}{2} =$ Mean speed to use in estimating revolutions.

CALCULATIONS FOR PITCH.

(21) T.S. = Tip speed for (12) (Table VIII).

(22) $1 - S = 1 -$ Basic Apparent Slip for S.B.C. and (12) (Table VIII).

(23) $P = \frac{\pi D \times V \times 101.33}{\text{T.S.} \times (1 - S)} =$ Pitch of Propeller in feet.

CALCULATIONS FOR REVOLUTIONS.

(24) Log A_v for V, (Table XIII).

(25) Log A_v for v, (Table XIII or XIV, depending upon type of vessels.).

(26) $s = S \frac{\text{I.H.P.}_a \times A_v}{\text{I.H.P.} \times A_v} =$ Apparent slip at speed v.

(27) $R_a = \frac{v \times 101.33}{P \times (1 - s)} =$ Revolutions for v.

Should the revolutions obtained be too great, at any value of e.h.p. ÷ E.H.P. on the Critical Thrust Curve of Fig. 4, it will be necessary to resort to one of the methods of *Reduced Diameter*, namely to design either for *Fan-Shaped* blades or for *Broad-Tipped* blades, using the first where the reduction in revolutions is considerable, and the second where this reduction is slight.

The forms for such computations are as follows:

	Fan-Shaped Blades.		Broad-Tipped Blades.	
Given	I.H.P.$_d$ R_d & v	e.h.p., R_d & v	I.H.P.$_{d1}$ R_d & v	e.h.p., R_d & v
(1) D = Diam. Max.	Constant	Constant	Constant	Constant
(2) D_1 = Diam > D	1-2-3	1-2-3	1-2-3	1-2-3
(3) $D_1 \div D^2$	1-2-3	1-2-3	1-2-3	1-2-3
(4) $(D_1 \div D)^2$			1-2-3	
(5) $(D_1 \div D)^{\frac{1}{2}}$	1-2-3	1-2-3		
(6) $(D_1 \div D)^{\frac{3}{2}}$				1-2-3
(7) $(D_1 \div D)^{\frac{1}{3}}$			1-2-3	1-2-3
(8) I.H.P.$_d$	1-2-3		1-2-3	
(9) I.H.P.$_{d1}$	I.H.P.$_d \div \frac{D_1}{D}$		I.H.P.$_d \times \left(\frac{D_1}{D}\right)^2$	
(10) ^sK = Thrust Deduction Factor	1-2-3		1-2-3	
(11) I.H.P.$_{p1}$ = I.H.P.$_{d1} \div K$	1-2-3		1-2-3	
(12) e.h.p.$_1 \div$ E.H.P. (as safe for S.B.C. and v (Table XVIII).				
(13) Z for e.h.p. $\div$ E.H.P.	1-2-3	1-2-3	1-2-3	1-2-3
(14) I.H.P.$_1$ = I.H.P.$_{p1} \times 10^z$	1-2-3		1-2-3	
(15) e.h.p. for v		1-2-3		1-2-3
(16) e.h.p.$_1$		e.h.p.		e.h.p. $\times \left(\frac{D_1}{D}\right)^{\frac{3}{2}}$
(17) E.H.P. = e.h.p.$_1 \div \frac{\text{e.h.p.}_1}{\text{E.H.P.}}$		1-2-3		1-2-3
(18) v = Des. Speed	Constant	Constant	Constant	Constant
(19) $v \div V$ (Fig. 4)	Constant	Constant	Constant	Constant
(20) $V = v \div \left(\frac{v}{V}\right)$	Constant	Constant	Constant	Constant
(21) I.T.$_p \div (1 - S)$	$\frac{C \times \text{I.H.P.}}{D_1^2 \times V}$		$\frac{C \times \text{I.H.P}}{D_1^2 \times V}$	
(22) E.T.$_p \times \frac{\text{P.A.}}{\text{D.A.}}$	$\frac{C \times \text{E.H.P.}}{D_1^2 \times V}$		$\frac{C \times \text{E.H.P.}}{D_1^2 \times V}$	

C = 3.84 for two-bladed screws; 2.88 for three-bladed screws; 2.491 for four-bladed screws.

(24) P.A. $\div$ D.A. for (21) (Table IX)	1-2-3		1-2-3	
(25) P.A. $\div$ D.A. for (22) (Table X)		1-2-3		1-2-3
(26) $\frac{1}{3}$ P.A. $\div$ D.A. for two-bladed screws	1-2-3	1-2-3	1-2-3	1-2-3
(27) $\frac{1}{4}$ P.A. $\div$ D.A. for four-bladed screws	1-2-3	1-2-3	1-2-3	1-2-3
(28) P.C. for (24) or (25) three-bladed, (26) two-bladed screws..., (27) four-bladed screws...	Table VIII for all columns.			
(29) E.H.P. = I.H.P. $\times$ (28)	1-2-3		1-2-3	
(30) e.h.p.$_1$ = E.H.P. $\times \frac{\text{e.h.p.}_1}{\text{E.H.P}}$	1-2-3		1-2-3	
(31) I.H.P.$_1$ = (17) $\div$ (28)		1-2-3		1-2-3
(32) I.H.P.$_{p1}$ = (31) $\div 10^{z(18)}$		1-2-3		1-2-3
(33) I.H.P.$_{d1}$ = (32) $\times$ (10)		1-2-3		1-2-3
(34) e.h.p. = e.h.p.$_1$ (30)	1-2-3		1-2-3	
(35) I.H.P.$_d$	I.H.P.$_{d1} \times \left(\frac{D_1}{D}\right)$		I.H.P.$_{d1} \div \left(\frac{D_1}{D}\right)^2$	

CALCULATION FOR PITCH.

	Fan-Shaped Blades.		Broad-Tipped Blades.	
(36) T.S. for (24) or (25)	1-2-3	1-2-3	1-2-3	1-2-3
(37) 1 − S for (24) or (25) and S.B.C.	1-2-3	1-2-3	1-2-3	1-2-3
(38) $P = \frac{\pi \times D_1 \times V \times 101.33}{T.S. \times (1-S)}$	1-2-3	1-2-3	1-2-3	1-2-3

CALCULATIONS FOR REVOLUTIONS.

(39) Log A_v for (20) (Table XIII)	1-2-3	1-2-3	1-2-3	1-2-3
(40) Log A_v for (18) (Table XIII or XIV)	1-2-3	1-2-3	1-2-3	1-2-3
(41) $s_1 = S \times \frac{I.H.P._{d1} \times A_v}{I.H.P._1 \times A_v}$	1-2-3	1-2-3	1-2-3	1-2-3
(42) $R_{d1} = \frac{v \times 101.33}{P \times (1-s)}$	1-2-3	1-2-3	1-2-3	1-2-3
(43) R_d	$R_{d1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{3}}$	$R_{d1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{3}}$		
(44) R_d			$R_{d1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{3}}$	$R_{d1} \times \left(\frac{D_1}{D}\right)^{\frac{1}{3}}$

In propellers of reduced diameter fitted to vessels of such type that the clearance between blade tips and hull has a governing effect on K, the value of K used should be that corresponding to the tip clearance of the Basic propeller or diameter D, and not the actual tip clearance. Reduced Diameter propellers enter and leave cavitation with the Basic propellers from which they are derived. The ratio of D to D_1 may be carried to as low as .80 with practical safety.

MATERIALS USED IN CONSTRUCTION OF PROPELLERS.

Blades: 1. *Cast iron:* Advantages: Cheapness, weakness which protects shafting by blade breaking upon striking obstruction.

Disadvantages: Heavy corrosion in salt water, heavy blade sections, blunt edges of blades.

2. *Semi-steel:* Advantages: Stronger than cast iron, cheapness.

Disadvantages: Liability to carry hidden flaws rendering it unreliable, corrosion in sea water.

3. *Cast-steel:* Advantages: Stronger than 1 or 2.

Disadvantages: Liability of hidden flaws, roughness of castings, corrosion in sea water.

4. *Forged-steel:* Advantages: Stronger and more reliable than 3.

Disadvantages: Expense of fabrication, corrosion in sea water.

5. *Manganese* and other *Strong Bronzes:* Advantages: Great strength, low ratio of thickness to width, fine sections, thin edges, high polish of surfaces, easily cast, practically free from corrosion.

Disadvantages: High first cost, liability to produce corrosion of adjacent parts of hull unless zinc plate protectors are fitted to the latter.

6. *Monel Metal:* Advantages: Great strength and toughness, light and sharp blade sections, high polish and non-corrosive in sea water.

Disadvantages: High first cost, difficult to machine, warps heavily in cooling in the mold, thus necessitating large amount of waste metal in rough casting in order to insure machining to designed pitch, tendency to porosity around edges of blades, liability to produce corrosion of adjacent parts of the hull unless zinc protector plates are fitted to the latter.

The Hub: Hubs are usually made of cast-iron or semi-steel for cast-iron, semi-steel and cast-steel built-up propellers; of semi-steel for the cheaper classes of work, and of manganese bronze for the better classes, with manganese bronze built-up propellers; of manganese bronze or Monel-metal for Monel-metal blades; where the propeller hub and blades are cast as one, of course the hub is of the same material as the blades.

THICKNESS OF PROPELLER BLADE AT ROOT.

T = Thickness of blade, in inches, at and tangent to the hub, additional thickness due to fillets being neglected. See Table XVI.

W = Width of blade tangent to hub, in inches.

A = (33,000 × I.H.P.$_d$) ÷ (2 π × R × Number of blades) = 5,252 I.H.P.$_d$ ÷ (R × N).

B = .31 × Diameter of Screw in feet = .31D.

C = A ÷ B.

D = 12 × B — Radius of hub, in inches.

E = C × D.

F = (33,000 × I.H.P.$_d$) ÷ (P × R × N).

G = .345 × Diameter of propeller in inches.

H = G — Radius of hub, in inches.

J = F × H.

K = Circumference of hub, in feet ÷ P, in feet.

L = Sine of Arc whose tangent is K.

M = Cosine Arc whose tangent is K.

N = L × J.

O = M × E.

e.h.p. ÷ I.H.P.	VALUES OF f			
	Constant Torque		Reciprocating Engines	
	Manganese Bronze	Cast Steel	Manganese Bronze	Cast Steel
.1	4800	4300	4800	4300
.2	5500	4900	5400	4850
.3	6300	5750	6000	5500
.4	7150	6650	6550	6050
.5	8000	7500	7150	6650
.6	9000	8450	7750	7250
.7	9800	9300	8300	7800
.8	10750	10250	8900	8400
.9	11600	11100	9450	8950
1.0	12500	12000	10000	9500

Table XIX. Values of f to be Used for Different Design Point Values of e.h.p. ÷ I.H.P.

$P = N \div O.$

f = fiber stress as per values of e.h.p. ÷ E.H.P. given in Table XIX.

$$T = \sqrt{\frac{P\ 13.125}{W \times f}}.$$

Fixing the maximum value of T at 2W, T should never exceed $T = \sqrt{\frac{P \times 2.625}{f}}$ for the strong bronzes.

For cast-iron and semi-steel, f = 2,500 to 4,000 for values of e.h.p. ÷ E.H.P. not in excess of .4.

CENTRIFUGAL FORCE; INCREASE OF STRESS.

W = Weight of blade, in pounds.

r = Distance of center of gravity of blade from center of hub.

g = Gravity = 32.

v = Velocity of center of gravity of blade, in feet per second.

$C = \frac{W}{g} \times v^2$, and the tension on the bolts = $\frac{W}{g} \times \frac{v^2}{r}$, $\frac{v^2}{r}$ being the accelerating force and called usually the centrifugal force.

When a propeller is in motion, with the ship advancing, running at R revolutions per minute, $v = \frac{R}{60}\sqrt{P^2 \times (2\pi r)^2}$.

To determine the bending moment at any distance d, from the axis, due to C, and resisted by the section at the root of the blade B.M. = $\frac{C}{d}$.

The bending moment in a plane at right angles to the axis of the screw at this distance d is

$$B.M. = \left(\frac{2\pi \times r \times 60}{R}\right)^2 \times \frac{W}{g} \times d.$$

DYNAMIC BALANCING.

By Commander F. J. Cleary, U. S. N., Member.

I have read Mr. Akimoff's article in the May number of the Journal of the A. S. N. E. with very great interest.

Since writing my original article, which appeared in the February issue, I have had the opportunity of testing under steam rotors that were first statically balanced only and then balanced dynamically by my method, and from the actual results obtained I know that the running conditions were greatly improved by such dynamic balance. I consider my method a satisfactory method. There are probably other methods as satisfactory as, or more satisfactory than, mine. If my work in this line arouses interest, as I believe it has, and through such interest other and better methods of balancing are developed, resulting in improved propelling machinery for the naval service and the merchant marine, I shall be quite content.

I am thoroughly familiar with the principles of the Carwen Dynamic Balancing Machine (Akimoff Patents), manufactured by The Carlson-Wenstrom Company, Philadelphia, Pa. The manufacturers claim that they can obtain in minutes a better dynamic balance than can be obtained in hours by other methods. If the quick results claimed are realized in actual practice, this machine would be unquestionably an improvement over the machine developed by me. The San Francisco Union Plant of the Bethlehem Shipbuilding Corporation have under purchase a balancing machine of the Carwen type (Akimoff Patents).

There are a number of points in Mr. Akimoff's discussion of my article on which I wish to comment. The quotations are from Mr. Akimoff's article in the May number of the A. S. N. E. Journal.

MR. AKIMOFF.—"The extract from the General Specifications of the U. S. Navy can only lead to confusion because, in the first place, no *perfect static balance can be obtained on knife edges;* and, second, because the operation under steam in its own casing may exhibit vibrations due to either static or to purely dynamic unbalance or to the mixture of the two; so that apparently suggesting a *method* of testing, this paragraph is absolutely misleading. The correct way would be to say: 'Great care will be taken to insure the rotor being balanced statically, the tolerance of the static unbalance being not over . . . inch ounce; then to balance the rotor dynamically, the tolerance of dynamic unbalance not exceeding . . . inch – ounce – inches.'"

As stated, the knife edge method is only an approximation. Consequently any static balance obtained by this method cannot be perfect. As a "method of testing," this paragraph in the General Specifications is believed to be entirely satisfactory, for if the rotor is run under steam in its own casing at a speed up to 25 per cent greater than the full-power speed without showing any apparent vibration, and a post trial examination shows no signs of rubbing, no loose blades, dummy packing, gland strips, etc., then it is absolutely certain that the unit will operate satisfactorily under service conditions as far as material is concerned. Mr. Akimoff's statement that standardized tolerances should be established for both static and dynamic unbalance is excellent and, in my opinion, should be embodied in the Bureau's specifications for all rotating elements.

MR. AKIMOFF.—"What is meant, probably, is the fact that all *mechanical* energy for rotating the body must come from *within* the oscillating system." "The flexible coupling is about the most unfortunate arrangement one could think of."

The statement in my original article was, "That the rotor must be so mounted that it can be rotated by an external applied force, but at the same time must be absolutely free from

any restraint or impulse from that external force." By the use of this flexible coupling results were obtained in which dynamic unbalance was eliminated to within negligible tolerances.

MR. AKIMOFF.—"It is not the existing vibration that should be magnified . . . there is no information that can be gained thereby."

The vibration was magnified for the same reason that a magnifying glass or a microscope is used,—to make the detection of defects easier.

MR. AKIMOFF.—"On page 7 dynamic balance is defined as a condition in which the centers of gravity of all elements lie on the axis of rotation. That this definition is wrong can instantly be seen by reference to a body such as a six-cylinder crankshaft."

Quite true; but my description dealt with the rotors, reduction gears and propellers of the *Shaw*, and the definitions were narrowed to the scope of the apparatus being described.

MR. AKIMOFF.—"All methods based upon the marking of 'high spots' are absolutely worthless; they are all based upon the value of the angle Σ of phase difference; now this angle depends upon the speed, ω, and therefore does not present anything tangible that can definitely be located. Reversing the rotation is likewise a questionable method, since it is not at all an easy matter to secure exactly the same speed as before. Furthermore, such a high spot, even if the speed could be accurately determined and maintained, would not be of any positive value that can be interpreted numerically."

By the use of an accurate tachometer and exact motor control there is no particular difficulty in securing and maintaining exactly the same number of revolutions in both directions of rotation, and under such conditions the marking of the "high spot" is absolutely accurate. The amount of horizontal displacement on the ground-glass screen is a positive measurement of the relative numerical value of the amount of unbalance before and after applying any trial-balance weight.

MR. AKIMOFF.—"I feel that the rotors (of the *Shaw*) are in neither static or dynamic balance at all."

The *Shaw's* turbines ran under steam in the shop test and on the ship's trials under the close scrutiny of the *Shaw's* officers and the Trial Board. The *Shaw's* first Commanding Officer and first Engineer Officer, who ran the *Shaw* over 60,000 miles, state that throughout the *Shaw's* turbines and propellers were in a state of practically perfect dynamic balance at all speeds.

MR. AKIMOFF.—"Also why the second propeller, not balanced statically, ran just as well as the other. Just think of a body balanced dynamically, but out of balance statically!"

In my article the statement was made that the second propeller, that was not balanced statically, was apparently as well balanced dynamically as the first propeller, that had been first balanced statically, *i.e.*, both propellers were in dynamic unbalance. Mr. Akimoff, I believe, misses the whole point of this phase of my article, which was that a careful static balance did not always aid in securing dynamic balance, but sometimes even increased the dynamic unbalance. In my method the static balance is secured while obtaining the dynamic balance. In other words, when a rotating symmetrical body is in practically perfect dynamic balance it is naturally in static balance as well.

The old adage that "the proof of the pudding lies in the eating thereof" can be invoked here. In addition to the *Shaw*, my method has been used with the propellers of the *Caldwell*, and with the propellers, the turbine rotors and the reduction gears of several other destroyers which have been built at this Yard. The only case in which wholly satisfactory dynamic balance was not obtained at all speeds during steaming trials was with one of the L.P. rotors of the Destroyer *Fairfax*. Due to non-arrival of a 450-H.P., 1,200 r.p.m. motor to be used in balancing, it was necessary to use a double-belt drive between a 400 r.p.m. motor and the rotor. Due to belt slip under the load, only 850 r.p.m. of the rotor could be obtained.

Under steam at 1,400 r.p.m. vibration occurred with this rotor which, while not excessive, was plainly noticeable.

It will be of interest to cite here some recent specific cases of dynamic balance secured at this Yard.

As an experiment, the starboard propeller of the U. S. Destroyer *Taylor* was carefully balanced statically on a fitted mandrel with journals six inches in diameter resting on two parallel knife edges ⅝ inch wide, by weights placed only on the studs in the centers of the blades. This propeller weighed 5,350 pounds. The static balancing was done with such accuracy that the propeller stayed in any position to which rotated (Fig. 1), and was so exact that a four-ounce weight hung on any stud would start the propeller rotating.

The center studs were 41 inches from center of hub, the side studs were 36 inches from center of hub (Fig. 3).

The propeller was placed on the dynamic balancing stand with the static weights secured in place on the center studs and rotated. Vibration began at 275 r.p.m., becoming violent at 325 r.p.m. By the addition of the following weights the propeller ran smoothly up to 500 r.p.m. (443 r.p.m. = 35 knots), No. 1 blade increased 3½ pounds to 4 pounds on center stud; No. 2 blade added 3 pounds on advance side stud, increased 6 pounds to 6½ pounds on center stud, added 1¼ pounds on following side stud; No. 3 blade added 4½ pounds on following side stud (Fig. 2). Metal was then chipped off as usual until propeller ran smoothly up to 500 r.p.m. with all weights removed.

The S. S. *Luckenbach* came to this Yard reporting that during the run from Seattle excessive vibration had developed in the port L.P. turbine (Parsons), and that later several rows of blades had been stripped in both rotor and casing. Whether the blade stripping was the cause of the excessive vibration or whether the excessive vibration caused the blade stripping, was not determined. The turbine was removed to the shop, opened up and examined. The rotor was found to be badly out of true and after considerable trouble was

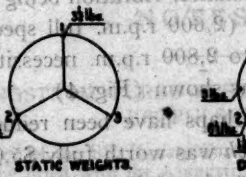


FIG. 1. FIG. 2.

STARBOARD PROPELLER
U.S.S. TAYLOR.

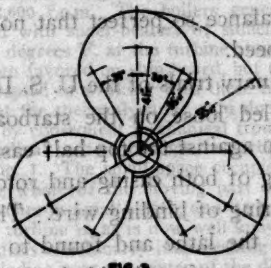
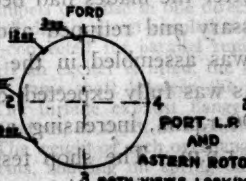
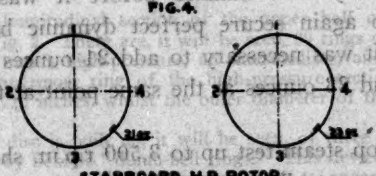


FIG. 3.



U.S.S. LUCKENBACK.

FIG. 4.



U.S.S. TAYLOR

FIG. 5.

straightened and rebladed. On the balancing stand it was found to be badly out of balance, vibration being violent at as low a speed as 525 r.p.m. (2,600 r.p.m. full speed). To secure dynamic balance up to 2,800 r.p.m. necessitated the addition of balance weights as shown (Fig. 4).

These weights could perhaps have been reduced or combined, but as the *Luckenbach* was worth fully $5,000.00 a day from a financial standpoint alone, the weights were secured, the turbine was assembled and placed aboard. Dock trials and a full-power run from Mare Island to San Francisco showed a dynamic balance so perfect that no vibration could be detected at any speed.

During the preliminary trials of the U. S. Destroyer *Taylor* the gland strips pulled loose on the starboard H.P. rotor, jamming the rotor up against the top half casing, resulting in a bad case of rubbing of both casing and rotor with deforming of blades and tearing of binding wire. The rotor was removed and placed in the lathe and found to be out of true. Simply as an experiment, after the blades had been straightened, renewed where necessary and retipped and the gland strips renewed, the rotor was assembled in the casing and given a shop steam test. As was fully expected, considerable vibration developed at 1,000 r.p.m., increasing as the speed was increased up to 1,500 r.p.m. The shop test was then stopped and the rotor placed on the balancing stand. (This rotor was in dynamic balance before it was first placed aboard.) To again secure perfect dynamic balance up to 3,300 r.p.m. it was necessary to add 21 ounces at one point at one end and 23 ounces at the same point at the other end (Fig. 5).

Another shop steam test up to 3,500 r.p.m. showed perfect dynamic balance at all speeds. The ship's trials (conducted up to 33 knots speed only, due to inexperience of personnel) showed no vibration at any speed.

NOTES.

THE LJUNGSTRÖM TURBO-ELECTRIC SYSTEM OF SHIP PROPULSION.

The machinery of the SS. *Wulsty Castle*, which is the first British sea-going ship to be fitted with the Ljungström turbo-electric system of propulsion, has been constructed by the Brush Electrical Engineering Company, Limited, of Loughborough, to the order of the British Ljungström Marine Turbine Company, Limited, of 22 Billiter street, London, E.C. The ship carries two Ljungström turbo-generators, each designed to develop 625 kw. when running at 3,600 r.p.m. The boilers generate steam at 220-pound gage, and are fitted with super-heaters of sufficient capacity to ensure a temperature of 625 degrees F. at the turbine stop valve. High superheats have not, so far, been favored on ship board, but the Ljungström turbine is particularly well adapted to working with these high superheats. There is thus no danger that the thermodynamic gain due to the high temperature will be discounted by consequent mechanical troubles.

A general longitudinal section through the turbine and its two generators is reproduced in Fig. 1. The small space occupied by the turbine itself is very striking. The whole of the moving part measures but 28 inches in diameter by 17½ inches in axial length, and its total weight is 4 cwt.

The Ljungström turbine is, as is now well known, a radial-flow machine. It consists of two discs which carry intermeshing rings of reaction blading. Steam enters the blading near the center of the discs, and as it passes from ring to ring, impels one disc in one direction and the other in the opposite direction. Each disc is direct-coupled to a generator, and the two generators are inter-connected for parallel running. Near the outer end of one generator shaft an exciter is fitted, the main governor and oil pumps are located on the same shaft between the exciter and the slip rings of the generator. The turbine exhaust flange is bolted direct on to the condenser, which serves, in short, as a general bed plate or foundation for the whole unit. The weight of each overhanging generator is borne by spherically-ended struts, abutting on spring buffers, as indicated in Fig. 1. By this arrangement, the condenser, turbine and generator casings are all free to expand with changes of temperature, and with a minimum of mutual distortion.

An enlarged longitudinal section of the upper half of the turbine is reproduced in Fig. 2. There are, it will be seen, 39 rings of blading, which are divided up into a high-pressure, an intermediate, and a low-pressure section. The innermost ring of the high-pressure section has an outer diameter of 3 1/16 inches, whilst the outer diameter of the ring No. 39 is 28 3/8 inches.

Each turbine disc is built up, it will be seen, of three components connected together by flexible "dumb-bell rings." This device, whilst securing accuracy of alignment, enables each component to expand freely without material constraint from the others. The innermost component of the three is exposed to high steam temperatures, whilst that carrying the low-pressure blading may have a mean temperature of perhaps not more than 150 degrees F. Serious expansion strains would accordingly arise were the disc solid, and the consequent distortion would make large clearances necessary, thus greatly augmenting leakage losses.

For similar reasons, each of the moving dummy plates *c*, Fig. 2, is connected with its turbine disc solely by two of the dumb-bell rings already described. All the steam that leaks through the dummy is necessarily highly superheated, and were part of the dummy solid with the turbine disc, the latter might be seriously distorted by having this high-temperature leakage steam on the one face and the lower-temperature "working" steam on the other. The methods of making the blading of the Ljungström turbine have already been described in detail in our issue of April 12, 1912. It will be sufficient therefore to recall here that the blades are machined out of rods of nickel steel. After cutting to length, tenons are machined on the ends of the blades. These tenons fit into holes punched in assembly rings of wrought iron. Even spacing and accuracy of angle is thus secured, and the whole is then welded up solid by the oxy-actylene process. The assembly rings are then turned to form, with a dovetail at the back, and this dovetail is secured by rolling into a corresponding slot formed in a stout carrier ring of high-tension steel.

In Fig. 2, the light assembly rings and the stouter carrier rings are clearly shown in the sections. These carrier rings are secured to the turbine discs by flexible dumb-bell rings of the type already described.

Owing to the care taken to minimize the distortion due to differences of temperature, very fine clearances are possible, though, of course, it is still necessary to follow Sir Charles Parson's maxim that where fine clearances have to be used between parts in rapid relative motion, one of the parts must be a thin fin only. If this precaution be adopted, a touch does no harm; the thin fin is not stiff enough to distort anything but itself, and though it gets hot, it simply rubs away, and provides its own clearance. In the Ljungström turbine the fins are made from strips of nickel steel, 5 mils. thick. These are secured by caulking into grooves formed in the blade rings, and are perhaps most clearly shown near the ends of the low-pressure rings in Fig. 2. Leakage by axial flow of the steam from between the blade rings has, in all cases, it will be observed, to make its way past two constrictions due to these rings, and the loss thus arising is, accordingly, very small. As will be seen from Fig. 2, the four last rings of blading are built up in three sections. This is primarily done to secure strength and stiffness, while at the same time increasing the turbine exhaust area. The load due to the centrifugal forces is thus carried by four rings of solid steel, well distributed, so that the distortion due to the centrifugal load is reduced to a minimum.

When the turbine is at work, the pressure tending to force the two discs apart axially is very considerable, amounting, in fact, to several tons. This end thrust is taken by the dummies shown at *cc*, Fig. 2. Each of these dummies consist of a pair of grooved discs, one of which is mounted on the turbine disc, and the other on the steam chest *d*, which is stationary. Steam flows through the space between the dummy discs, past numerous baffles, and finally escapes into the exhaust chest of the turbine. The arrangement of the baffles is shown diagramatically in Fig. 4. Here A represents part of the rotating dummy disc, and B part of the stationary disc. As will be seen, fins project from one disc into grooves cut in the other. Each of the fins is provided at its outer end with a baffle plate of nickel steel 5 mils. thick. The edges of this plate just clear by some 5 mils. or so the adjacent edges of the groove with which they are meshed, and through these narrow inter-spaces the steam must find its way in leaking out to the exhaust.

As shown in Fig. 4, two forms of groove are used in the dummy, the object being to secure automatically an exact balance of the axial thrust, tending to separate the main turbine discs. As represented in Fig. 4, all the baffles are effective, the steam being as effectively wire-drawn at *aa* as at *bb*. This, however, is not, save by accident, the working position of the dummy discs relatively to each other. Let it be assumed, for example, that

the end thrust is exactly balanced in the position shown when the vacuum is 27 inches. Suppose now that the vacuum is increased to 28 inches. Such an increase as is well known will hardly affect more than the last row or two of the turbine blading, so that the total end thrust tending to separate the turbine discs is changed very little. The pressure in the exhaust space, which tends to force the two discs together, has, however, been reduced. Hence, there is now an unbalanced pressure tending to separate the turbine discs. These, therefore, move apart, carrying with them their dummy discs. The effect of this motion is that the baffles at *aa*, Fig. 4, are carried into the widened spaces or pockets shown, thus greatly increasing the clearance here, whilst the clearances at the grooves *bb* are unaffected. The consequence is that to get the same weight of steam past the *b* grooves as now passes the *a* grooves, the pressure at the commencement of the "*b*" section of the dummy must rise. The general result is that the total pressure tending to separate the components of the dummy is increased; thus correcting the "out of balance" of the turbine discs due to the change in the vacuum. In the contrary case, should the vacuum diminish from the original standard of, say, 27 inches, the balance between the dummies and turbine discs will be disturbed in the opposite direction. The two turbine discs will then tend to close in towards each other. In moving thus, however, the baffle strips at the *b* grooves of the dummy are, in their turn, carried into widened pockets. Hence the steam which has leaked past the *a* grooves now gets away more freely than before, and the total pressure tending to separate the dummies is decreased, thus again correcting the "out of balance."

To prevent hunting, a dashpot is provided at the outer end of each generator shaft. One of these dashpots *h* is shown in position on the extreme right of Fig. 1, and in detail in Fig. 7. It consists essentially of two pistons *i i* on opposite sides of a diaphragm *k*. The two pistons are coupled to the end of the generator shaft by double-thrust ball-bearings, as shown. The space between each of the pistons and the diaphragm is connected up through a non-return ball valve, with the general oil supply of the turbine. An adjustable leak-off, connecting the two sides of the diaphragm is provided at *j*. Any tendency to hunt is damped out by friction at this plug.

As already mentioned, the steam enters the turbine blading near the center, and flows radially outwards. Were the working agent water, the simplest method of conveying it to the center of the discs would, no doubt, be to use hollow shafting, through which the fluid would be passed. Such a scheme is impracticable, however, when the working agent is highly superheated steam. Bearing troubles would probably be serious, and the heat which would with this arrangement pass from the steam to the generator rotors would not be welcomed by the designer of the electrical section of the plant. Another plan was, therefore, necessary, and the turbine is fitted accordingly with two steam chests *d d*, Fig. 2. These are ring-shaped steel forgings, which at the one end support the gland bushing, and at the other the standing half of the dummy. To ensure against distortion by expansion strains, these chests are each, it will be seen, connected to the casing by one of the dumb-bell shaped rings already referred to.

Steam is led into these chests by a breeches pipe *p p*, of which one section is shown below the turbine in Fig. 1, whilst a bastard cross-section, showing the connection to the main steam stop-valve, is reproduced in Fig. 3. The steam chest *d* is, it will be seen, double walled, and the steam pipe passes through both walls into the inner section, which surrounds the shaft, as best seen in Fig. 2. If the turbine is intended to take an overload, a special valve is fitted, which admits steam into the outer annulus of the chest, from which it enters the turbine through the by-pass openings shown at *k k* in Fig. 2. In the present case no overload valve is fitted, but a pipe not shown in the figure allows steam to be drawn off from this annulus at

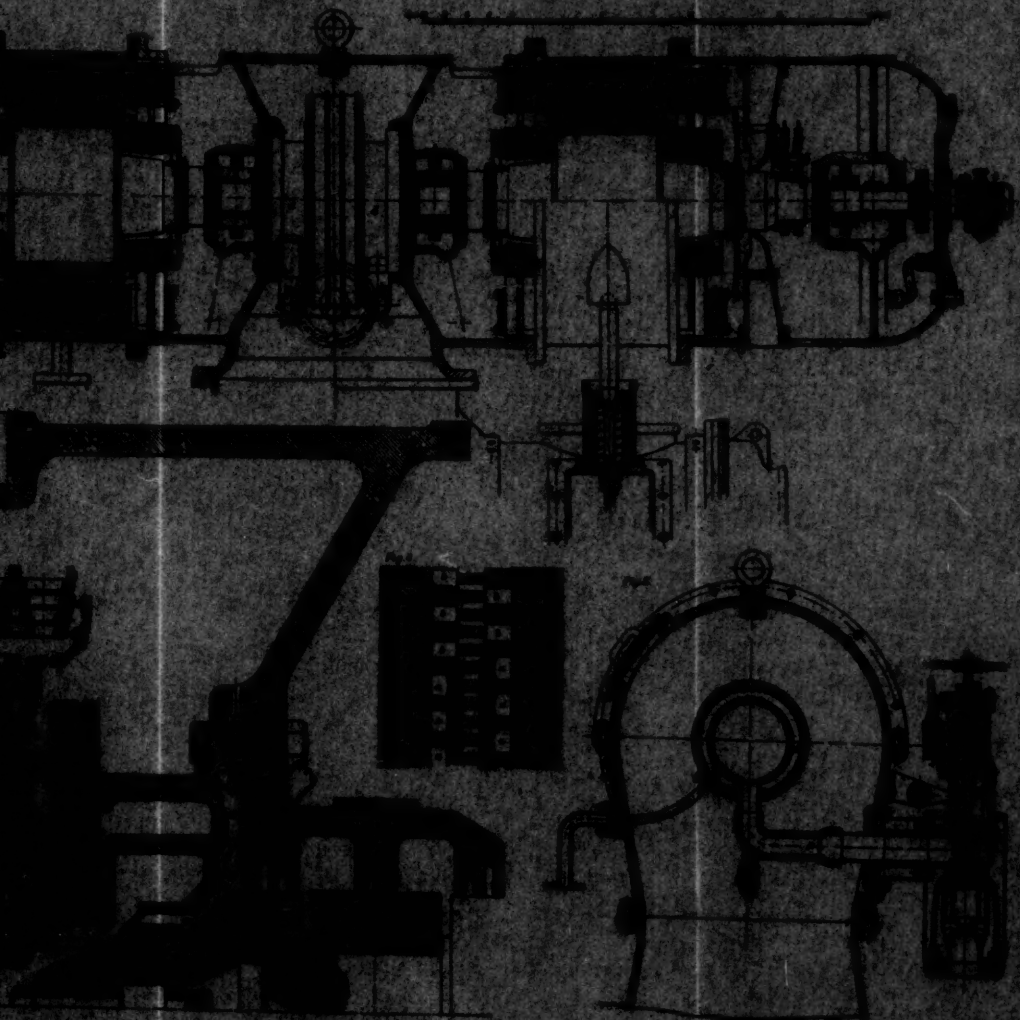
about 40 pounds' pressure for feed-heating purposes. The pipe shown to the left does not connect with the steam chest at all, the section being a bastard one, but traps the leakage from the glands. To reduce radiation and convection losses, the exposed outer surfaces of the steam chest are all jacketed, as indicated at *y*, Fig. 2.

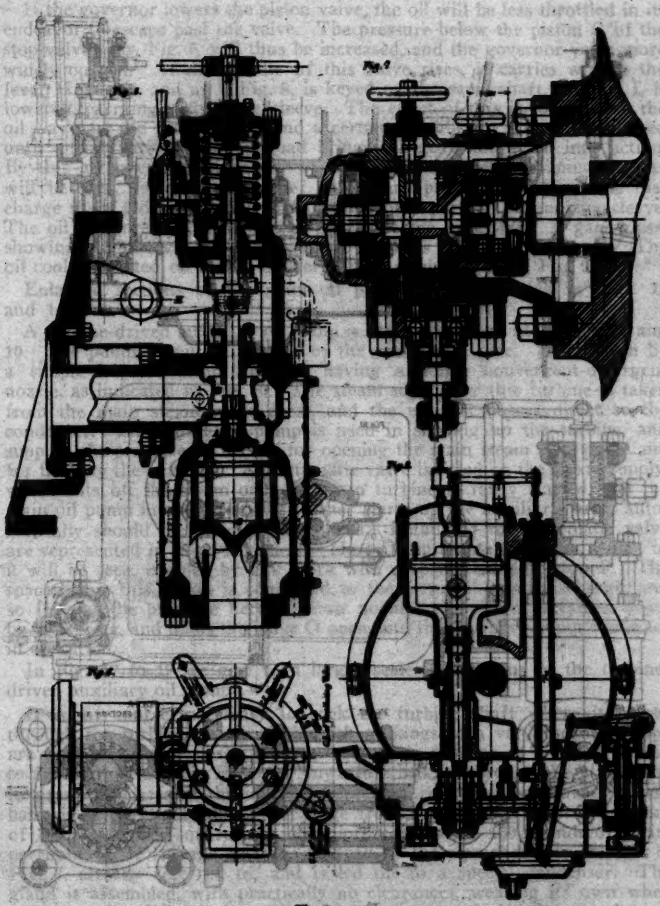
Each turbine disc is mounted on a conical seat, formed on the end of a short stub end of hollow shafting *s*, and is secured there by four conical pins, which are held in place by the plug *w*. The stub shaft, in its turn, is registered into the end of the generator shafting, and secured in place by the bolts *t*. To dismantle the turbine, the upper half of the casing is removed, and also the top caps *u* of the bearings, which are lifted by a shackle screwed into the tapped hole shown. The bolts *t* can then be removed. The two turbine elements are next clamped together in a special fixture, which holds them in true alignment with each other, and the dummies. The generator shafts are then withdrawn sufficiently to clear the registering spigot on the stub shaft, and the turbine can then be lifted out complete and dismantled in detail if required. The bolts *t* are knurled in the body, as indicated, and when home, soft metal is caulked down on to the knurling, thus effectively locking the bolts in place.

As will be seen, oil baffles are provided at *x* to prevent the escape of lubricant from the bearing casings.

The steam stop-valve is combined in one casing with the governor valve, and is shown in position to the right of Fig. 3, whilst Figs. 5 and 6 shows the whole arrangement in detail, and to a larger scale. The stop valve A, Fig. 5, is of the ordinary pot-lid type, and is surmounted by a balanced double-beat governor valve B, secured to the same spindle. This valve B is designed so that it does not open appreciably until A is well clear of its seat. As a consequence, B may be moved through a considerable range of opening without sensibly affecting the steam way past A. The governor valve is operated by oil pressure, which is admitted below the spring-loaded plunger C. The pressure of this oil supply is regulated by the governor, as will be described later on. As the speed rises, the governor reduces the oil pressure, so that the spring moves down C, and with it the governor valve B, thus throttling the steam supply. The screwed spindle D is simply used to lock home the valve A. It can be used to close this valve, but not to open it. Hence, in starting up the turbine, the procedure is to run up D, thus unlocking the valves. The latter, however, cannot yet open, because the spring above C is sufficiently powerful to hold A closed against the full steam pressure below it. On raising, however, the pressure in the oil circuit, either by a hand pump or by an auxiliary steam pump, the piston C is lifted, opening the valves and admitting steam to the turbine. Since the one oil circuit supplies both this piston and the bearings, the arrangement just described ensures that the turbine cannot be started up until the bearings are thoroughly flushed.

The valve shown at E, Fig. 5, is an equilibrium valve. In the position shown, it admits oil below the piston C only. Should, however, this valve E be raised, it establishes equilibrium between the upper and lower sides of C, which is then forced down by the spring, and closes the main stop valve A. The valve E is, it will be seen, provided with a spring which constantly tends to raise it into the equilibrium position. This is resisted by the stop F, and if this stop be thrown over by the handle or otherwise, the turbine is forthwith shut down. A cable passes through the upper hole shown in F, and so long as it is taut, the stop is held in the position shown, and the valve E remains in its lowest position. This cable is connected to the emergency governors (of which there are two, one for each generator shaft) in such a way that if either governor acts, the cable is slacked off. The spring is then able to push over the stop F and move up E.

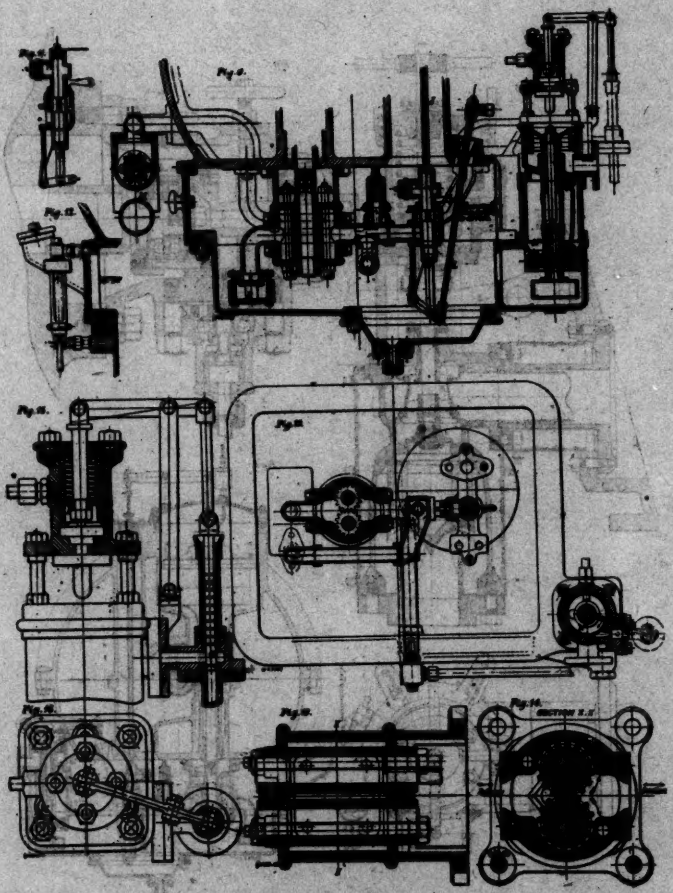




FIGS. 5–8.

A strainer G is fitted below the stop valve A to prevent loose objects accidentally left in the steam pipe being carried into the turbine.

As already mentioned, one exciter is used for both generators, and is arranged in the left-hand side of the set as indicated in Fig. 1. Between it and the generator, the governor, oil pumps and oil tanks are situated. A cross-section through these is reproduced in Fig. 8. The governor, it will be seen, is driven by a spiral gearing. The main oil pump, which is of the gear type, takes its drive from the bottom of the governor spindle. The weight of the latter is carried on a ball thrust, as shown.



FIGS. 9–16.

The lever I, shown to the left of Fig. 8, is keyed to the same shaft as the lever H, Fig. 5. It operates the "overtaking gear" by means of the link work shown. The governor, as the speed varies, raises or lowers the rod J, and with it the piston valve, which controls the supply of oil to the piston C, Fig. 5. The arrangement of this valve is shown to a larger scale in Figs. 9, 10, and 11. As there shown, the piston valve operated by J moves in a sleeve coupled up to the overtaking gear. The oil supplied by the pump finds its way past the piston valve into the chamber K, which is coupled up to the oil supply main.

If the governor lowers the piston valve, the oil will be less throttled in its endeavor to escape past the valve. The pressure below the piston C of the stop-valve gear, Fig. 5, will thus be increased, and the governor valve more widely opened. As the spindle of this valve rises, it carries with it the lever H, Fig. 5, and as I, Fig. 8, is keyed on the same shaft, the link L is lowered, carrying with it the sleeve. The effect of this is to diminish the oil way past the piston valve, and a certain equilibrium is thus established until a new variation in the load brings the governor again into action. By this arrangement all danger of hunting is completely eliminated. As will be seen, spring-loaded relief valves are provided, both on the discharge main from the pump and on the upper part of the moving sleeve. The oil pressure can, accordingly, never become excessive. A gage-glass, showing the oil level in the tank is provided, as indicated in Fig. 12. The oil cooler is fitted on the oil-delivery main, as shown at M, Fig. 9.

Enlarged sections of the main oil pump are represented in Figs. 13 and 14.

A turbine-driven auxiliary oil pump is fitted, as shown at N, Figs. 9 and 10. The pump, it will be seen, is of the centrifugal type. It is driven by a simple impulse steam turbine, having a single convergent-divergent nozzle, as indicated in Fig. 10. The steam supply for this turbine is taken from the main steam valve chest, and the exhaust passes direct to the condenser. This auxiliary pump is used in starting up the turbine, and supplies the pressure oil needed for opening the main steam stop valve, and for flushing the bearings. An automatic valve is fitted on the steam supply, which cuts off the steam once the main turbine is fairly started, and the main oil pump in operation. It further starts up the auxiliary pump automatically should the main oil pump fail. Details of this automatic valve are represented in Figs. 15 and 16. Here O represents the valve, which is, it will be seen, coupled by link work with a spring-loaded piston P. The space below this piston is connected to the main oil-pump discharge, and so long as the pressure here is below normal, the piston is forced down by the spring, and holds the valve O open, and the auxiliary pump continues in operation.

In Figs 17 to 19 we give to a larger scale illustrations of the turbine-driven auxiliary oil pump

It is, of course, necessary to pack the turbine shaft where it passes through the turbine casing. Labyrinth packings of a very ingenious type are used for this purpose. They are shown in position at *v v*, Fig. 2, and to an enlarged scale in Fig. 20. Any steam leaking through the glands has to pass in succession through the numerous constrictions shown. These baffle it so effectually that the leakage from both glands under a pressure of 180 pounds per square inch does not exceed some 150 pounds of steam per hour. The leakage steam is trapped by the pipe shown to the left in Fig. 3, already referred to, and is led off to a special condenser. The gland is assembled, with practically no clearances, wearing its own when set to work. Where they touch, the fine ends of the fins simply rub away, their mass being too small to occasion any distortion of the gland by the heat developed in the process.

In Figs. 21, 22 and 23, we show some further details of the Y pipe, by which the steam passes from the main stop valve to the turbine valve chests. This pipe has to pass through the exhaust space, and to reduce convection losses, it is air-jacketed. The jacket is of copper made in halves which were brazed together after being placed on the main pipe. At the two ends of the jacket the metal is turned inwards, so as to grip the pipe and cause the jacket to take up a position approximately concentric to the pipe it covers. This feature is shown clearly in the detailed sketch of one end of the jacket annexed to Fig. 22. We may add that the

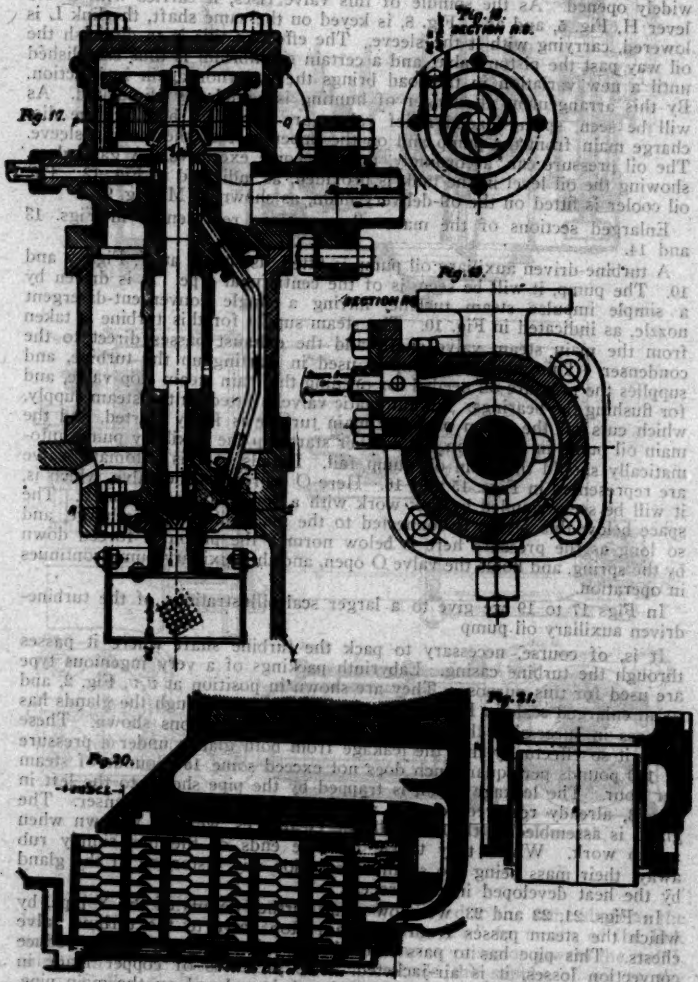
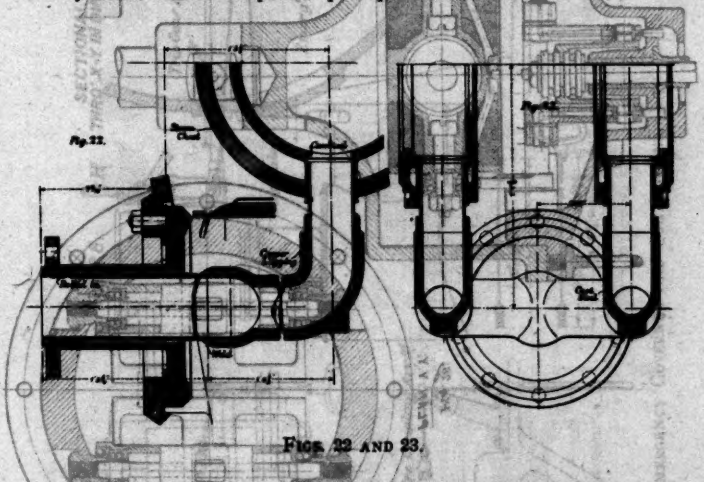


Fig. 17.
Fig. 18.
SECTION B-B.
Fig. 19.
SECTION
Fig. 20.
Fig. 21.

word "weld" on the horizontal part of the pipe in this illustration refers to the union between the cast-steel portion of the pipe and the horizontal solid-drawn steel tube. The ends of the Y pipe are as shown in Figs. 22 and 23, and to a larger scale in Fig. 21, so made as to be a sliding fit into steel sleeves, which are caulked into the steam chest. Owing to this arrangement, there are no pipe connections to be dismantled in removing the turbine for examination or repair. The pipe work here was subjected to a hydraulic test of 400 pounds per square inch.

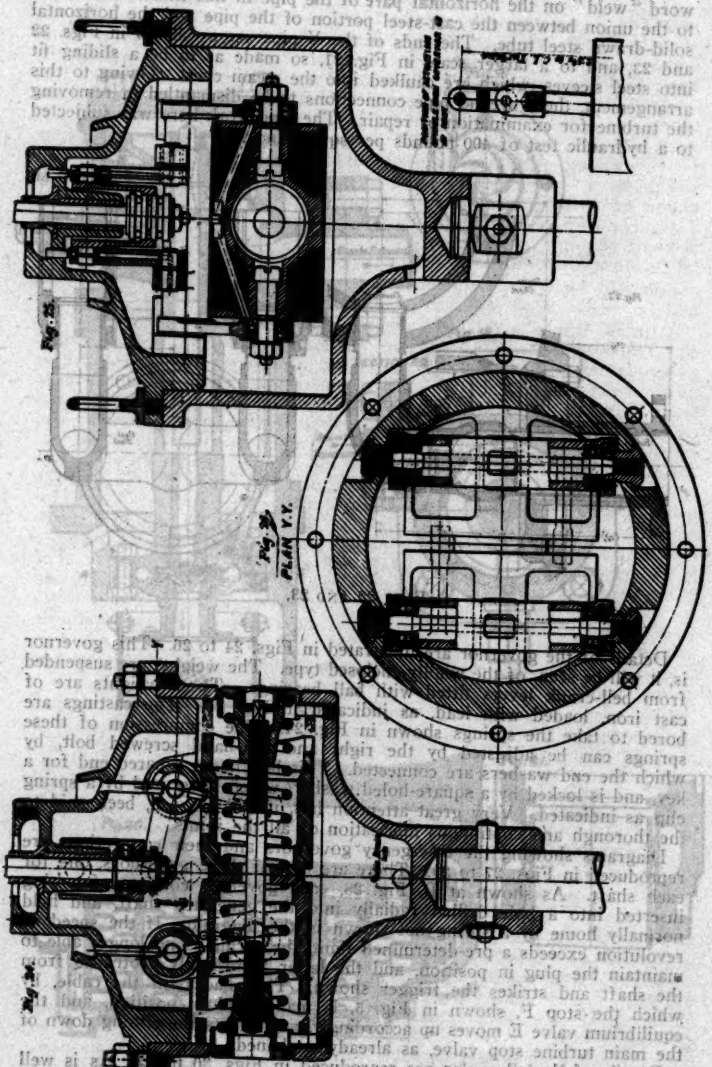


FIGS. 22 AND 23.

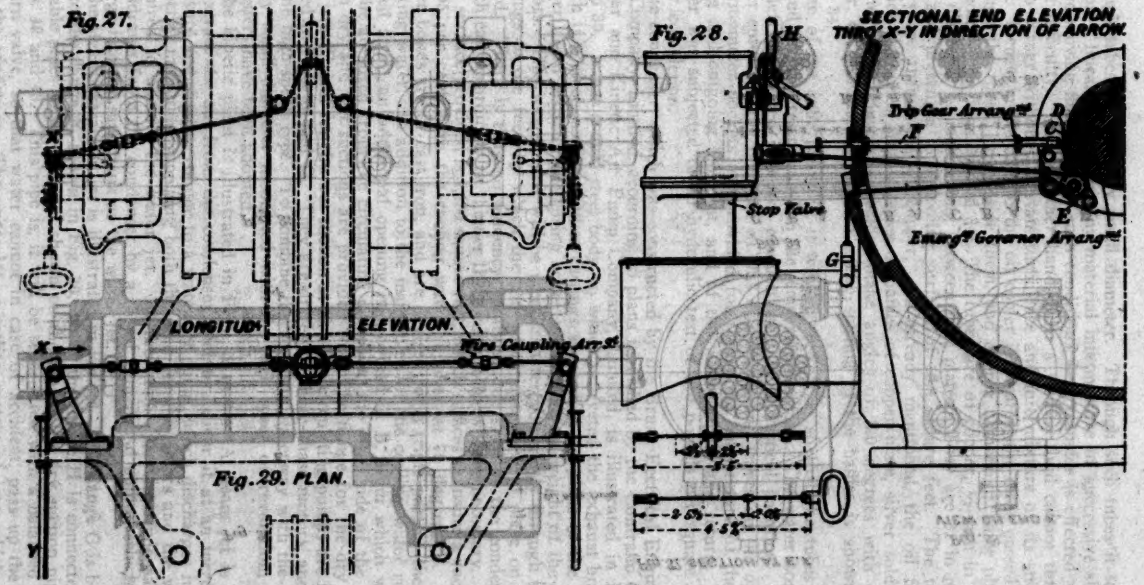
Details of the governor are illustrated in Figs. 24 to 26. This governor is, it will be seen, of the wholly-enclosed type. The weights are suspended from bell-crank levers fitted with ball bearings. These weights are of cast iron, loaded with lead, as indicated in Fig. 25. The castings are bored to take the springs shown in Fig. 24. The compression of these springs can be adjusted by the right and left-hand screwed bolt, by which the end washers are connected. This bolt has a squared end for a key, and is locked by a square-holed washer, which is secured by a spring clip as indicated. Very great attention has, it will be seen, been paid to the thorough and continuous lubrication of all the joints.

Diagrams showing the emergency governors and their connections are reproduced in Figs. 27 to 29. There are two of these governors, one for each shaft. As shown at B, Fig. 28, each governor consists of a plug inserted into a hole drilled radially into the generator shaft, and held normally home by a spring not shown in the diagram. If the speed of revolution exceeds a pre-determined limit, the spring is no longer able to maintain the plug in position, and the latter accordingly protrudes from the shaft and strikes the trigger shown. This slacks off the cable, by which the stop F, shown in Fig. 5, is maintained in position, and the equilibrium valve E moves up accordingly, and effects the closing down of the main turbine stop valve, as already explained.

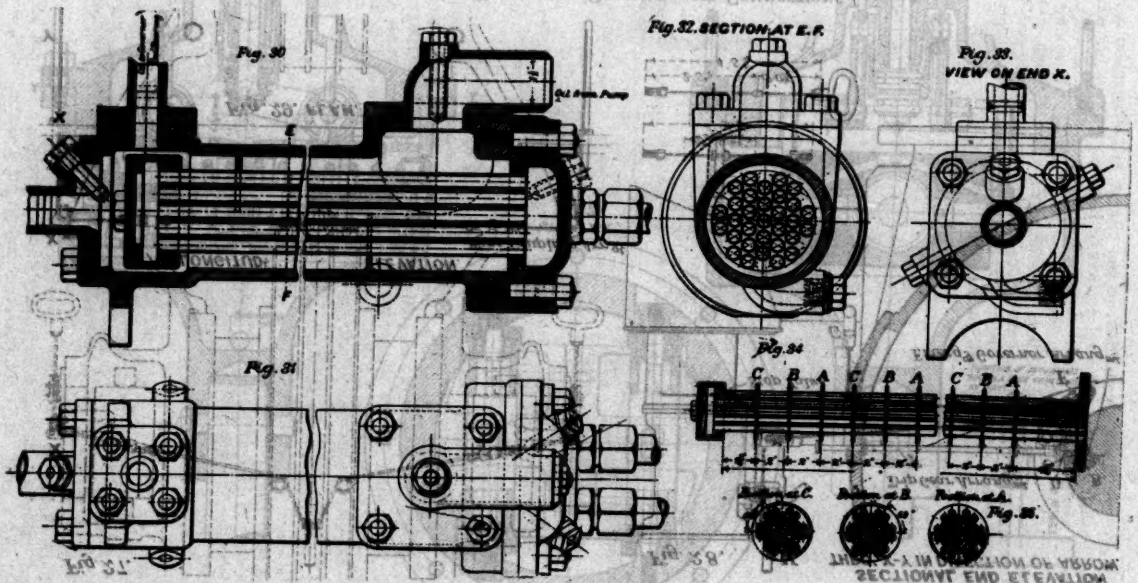
Details of the oil cooler are reproduced in Figs. 30 to 33. As is well known, turbulence is necessary to any rapid interchange of heat between a fluid and a solid. It is not, however, easy to get a viscous fluid, such as lubricating oil, into a condition of turbulent flow, especially if it moves



FIGS. 24 TO 26.—GOVERNOR DETAILS.



FIGS. 27 TO 29.—EMERGENCY GOVERNOR AND CONNECTIONS.



FIGS. 30 TO 35.—OIL COOLER.

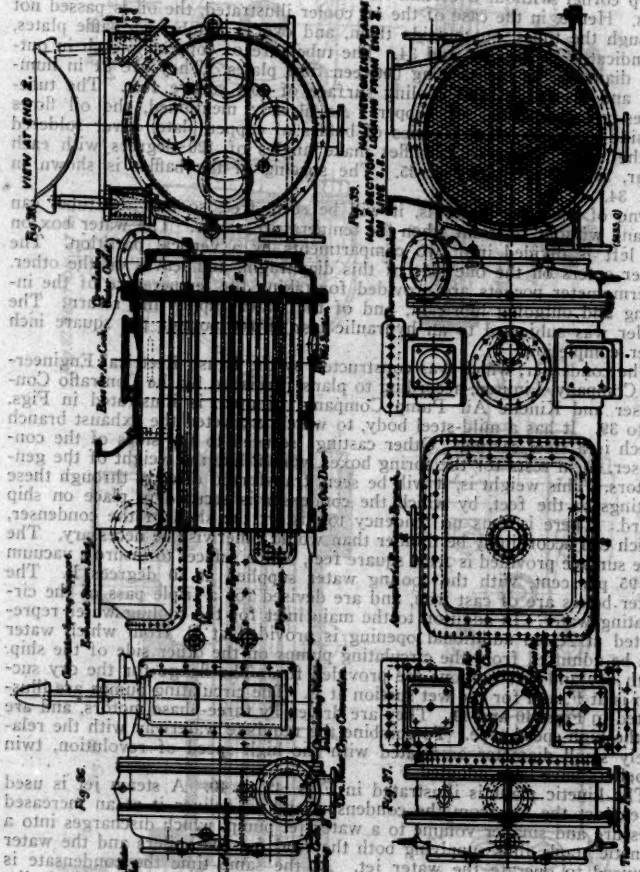
through tubes of relatively small diameter. Through such tubes it tends to flow "linearly," without any material intermixture of successive shells of the moving fluid. Under such conditions, heat transfer is effected with great difficulty. On the other hand, even a viscous fluid cannot turn a sharp corner without a certain turbulence and intermixture of its stream lines. Hence, in the case of the oil cooler illustrated, the oil is passed not through the tubes, but outside them, and over a number of baffle plates, as indicated in Figs. 30 and 34. The tubes are of copper, ½-inch in outside diameter, 34 inches long between tube plates. They are 31 in number, and present a total cooling surface of 10.8 square feet. The tube-plates are themselves of copper. As already mentioned, the oil flows outside the tubes, past a series of baffles of copper plate, silver soldered to the tubes. Successive baffles make angles of 120 degrees with each other, as indicated in Fig. 35. The spacing of the baffles is shown in Fig. 34.

One of the tube plates is, it will be seen, free, so that the tubes can expand without restraint when the temperature varies. The water box on the left is divided into two compartments by a vertical partition. The water enters on the one side of this diaphragm, and leaves on the other. Thermometer pockets are provided for taking the temperature of the ingoing and outgoing lubricant, and of the water supply and return. The cooler was subjected to an hydraulic test of 140 pounds per square inch after completion.

The condenser, which was constructed by the Brush Electrical Engineering Company, Limited, according to plans furnished by the Contraflo Condenser and Kinetic Air Pump Company, Limited, is illustrated in Figs. 36 to 39. It has a mild-steel body, to which is riveted the exhaust branch which is a steel casting. Other castings secured to the sides of the condenser, form rests for the spring boxes, which take the weight of the generators. This weight is, it will be seen, transferred directly through these castings to the feet, by which the condenser is secured in place on ship board. There is thus no tendency to distort the shell of the condenser, which can accordingly be thinner than would otherwise be necessary. The tube surface provided is 1,150 square feet, which suffices to insure a vacuum of 95 per cent, with the cooling water supplied at 70 degrees F. The water boxes are of cast iron, and are devised for a triple pass of the circulating water. In addition to the main inlet for the cooling water represented at A, an additional opening is provided at B, from which water can be admitted from the circulating pumps on the other side of the ship. Both dry and wet suctions are provided for. The flange for the dry suction is at F, and for the wet suction at E. The circulating pumps are illustrated in Figs. 40 and 41. They are driven by three-phase motors, and are of the high-speed type. To combine the requisite waterway with the relatively small diameter associated with the high speed of revolution, twin impellers have been adopted.

The kinetic plant is illustrated in Figs. 42 to 46. A steam jet is used to extract the air from the condenser, and to deliver it at an increased pressure and smaller volume to a water jet pump, which discharges into a "kinetic" tank, thus supplying both the feed for the boilers and the water required to operate the water jet. At the same time the condensate is withdrawn from the condenser by a set of centrifugal pumps, which discharge into the same tank as the water jet.

The steam extraction jet is illustrated in Fig. 46. The flange G is bolted on to the flange F, shown in Fig. 36, whilst the flange H is connected by suitable pipes to the flange I shown on the top of the dry suction pipe in Figs. 42 and 43. This pipe is, it will be seen, fitted with a balanced non-return valve, so that water cannot in case of accident, pass up the pipe into the condenser. The water jet or rather, system of jets, which complete the compression and delivery of the air is shown at K, Fig. 42. The



Figs. 26 to 28.—Condenser Constructed from the Designs of the Contarto Condenser and Kinetic Air Pump Company, Ltd.

air delivered past the non-return valve already mentioned, is impelled by the inductive action of the small central jet into the large nozzle shown. A further supply of water is admitted through the annulus round this large nozzle, and the mixed contents of air and water pass on through the diffusing pipe into the kinetic tank by the opening shown at L. The area through the annulus round the large nozzle is adjustable by means of the screw seen at M, Fig. 47, which shows the details to a larger scale.

The feed for the system of jets described is withdrawn from the kinetic tank by a motor-driven centrifugal pump, the discharge pipe from which is shown at N. The water passes through the strainer O, Figs. 42 and 43, and thence on to the jets as indicated. The whole of the water does not, however, pass through the jets. Part of it is delivered by the branch P, Fig. 42, and the float regulated valve Q, Fig. 45, to the space below the non-return valve R, and thence to the feed tank or hot well. The non-return valve and its connections is shown more clearly in the larger scale views, Figs. 48 and 49. On its passage from this valve to the hot well, this water passes through a small independent surface condenser, shown at S, Fig. 42. To this condenser the steam that leaks through the turbine glands is led, as already stated. This steam, by its condensation, heats the water on its way to the feed tank.

The float-regulated valve Q is intended to prevent the possibility of the water level in the kinetic tank falling below the inlet to the kinetic pump. Should this limit be approached, the falling float moves over the valve Q, so as to cut off the flow to the feed tank. The float T is hollow, and is kept constantly full of water by a discharge from the pipe U, which is coupled with the pipe P. This arrangement has been adopted because hollow floats having their interiors sealed off, have at times given trouble by proving porous, so that it was considered preferable to waterlog the float once and for all. A counter weight is then, of course, necessary, and is provided as shown. The pumps used are all arranged on one vertical shaft, and are driven by the three-phase motor shown in Fig. 42, and perhaps still better in the general view of the plant reproduced in Fig. 57.

The pumps for extracting the condensate are compounded. One pump serves merely to extract the condensate and to maintain a certain head on the suction of its successor, by which the bulk of the work is done.

A large scale section through one of the generators is represented in Fig. 50. Each generator is designed to develop 312½ kw. at 650 volts when running at 3,600 r.p.m. The cooling of both stator and rotor is provided for by a single fan, the arrangement being such that all interspaces can be cleaned without stripping the machine. The rotors are solid steel forgings with milled slots, which take former-wound coils. These are held in place by gun-metal wedges. Special gun-metal caps are fitted over the end windings to secure them against displacement by the heavy centrifugal forces.

The current from each turbine unit is employed to actuate an induction motor, a section through which is shown in Fig. 51, page 832, and both these motors drive, by double-helical gearing, the propeller shaft. A view of these two motors in course of erection at the Falcon Works, is reproduced in Fig. 52, page 832, whilst Fig. 53 is a view showing the stators and part of the flexible couplings by which connection is made to the pinion shafts. Curves showing the results obtained on test are reproduced in Fig. 54. It will be seen that an efficiency of 95 per cent was realized with a power factor of .875. The normal running speed for these motors is 714 r.p.m. This is reduced by the gearing to the 76 turns per minute required for the propeller shaft, which at this speed absorbs 1,500 s.h.p. The gearing, which was supplied by The Power Plant Company, Limited, West Drayton, shown in photographs, Figs. 55 and 56. The pinions are 9.3 inches in diameter at the pitch line, and have 23 teeth. The wheel with which they gear is 86 inches in diameter, and the reduction ratio is 9.4 to 1.

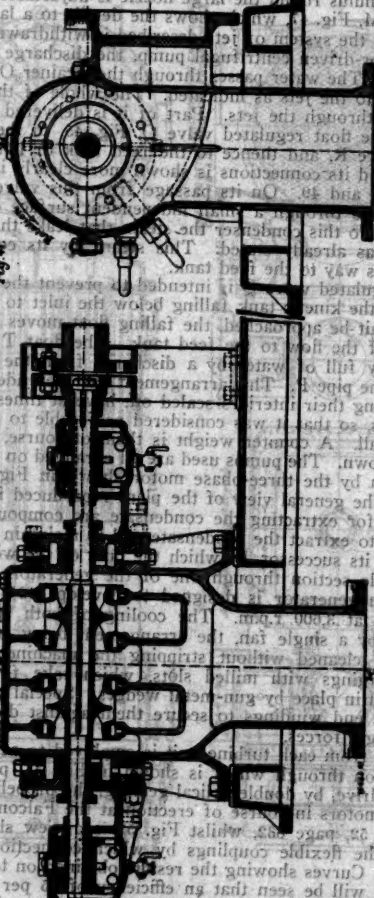


Fig. 49.—Section E.E.
Fig. 50.—Section F.F.

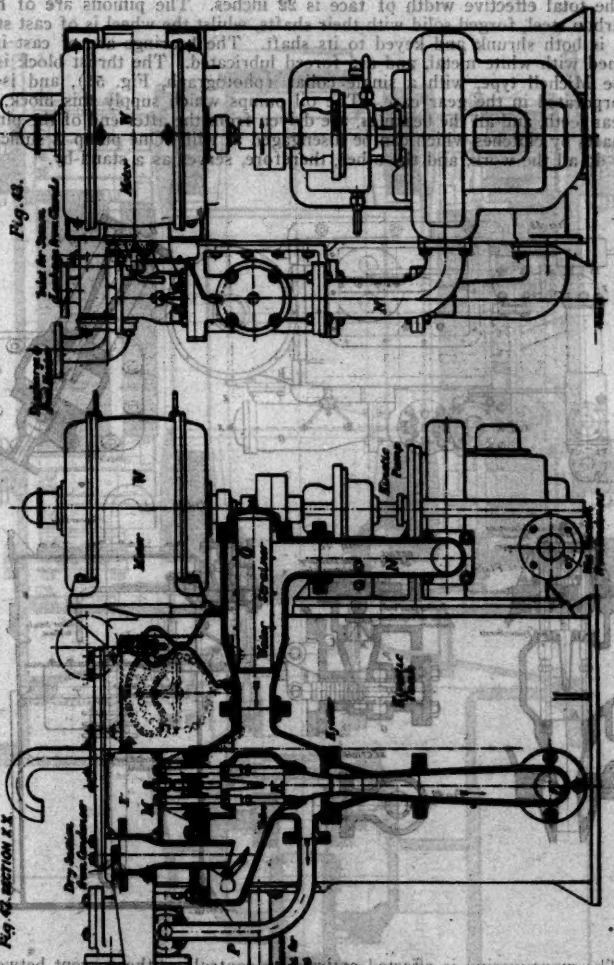
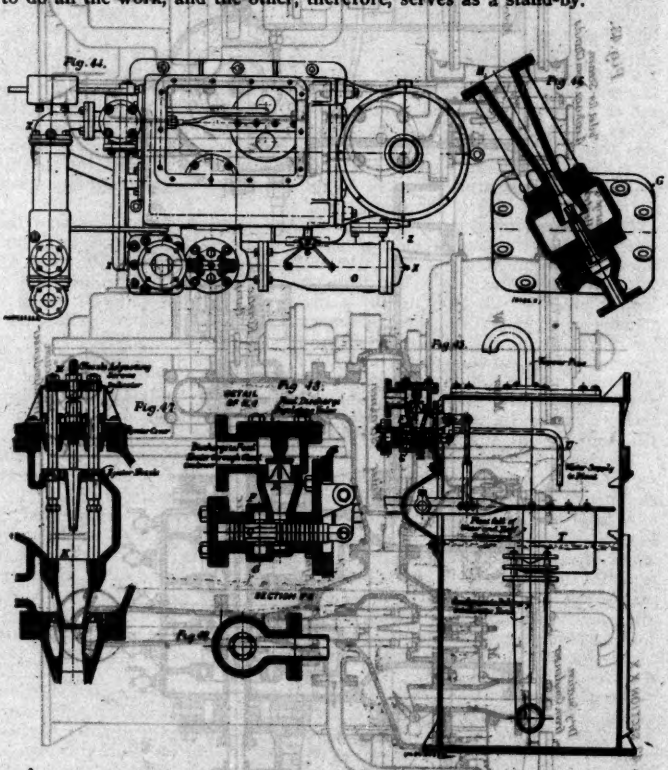


Fig. 61. SECTION E.K.

The total effective width of face is 22 inches. The pinions are of high carbon steel, forged solid with their shafts, whilst the wheel is of cast steel. It is both shrunk and keyed to its shaft. The bearings are of cast iron, lined with white metal, and are forced lubricated. The thrust block is of the Michell type, with a single collar (photograph, Fig. 56), and is incorporated in the gear case. The oil pumps which supply this block, the gear teeth, and all the bearings, are driven from the after end of the pinion shafts by clutches which can be disengaged at will. One pump is sufficient to do all the work, and the other, therefore, serves as a stand-by.



The maneuvering is effected entirely by controlling the current between the generators and the motors. The gear is illustrated in Figs. 58 to 60, and also in Fig. 61. The amount of current is controlled by liquid resistances, the electrolyte being a solution of KOH. The electrodes have the form of cones, as shown in Fig. 58. The lower cones are stationary and filled with the solution, whilst the upper ones are mounted on spindles which are raised or lowered by means of a hand wheel shown in Figs. 59 and 60. In the stop position of the hand wheel the upper cones are lifted entirely out of the liquid in the lower cones, whereby the circuit is broken.

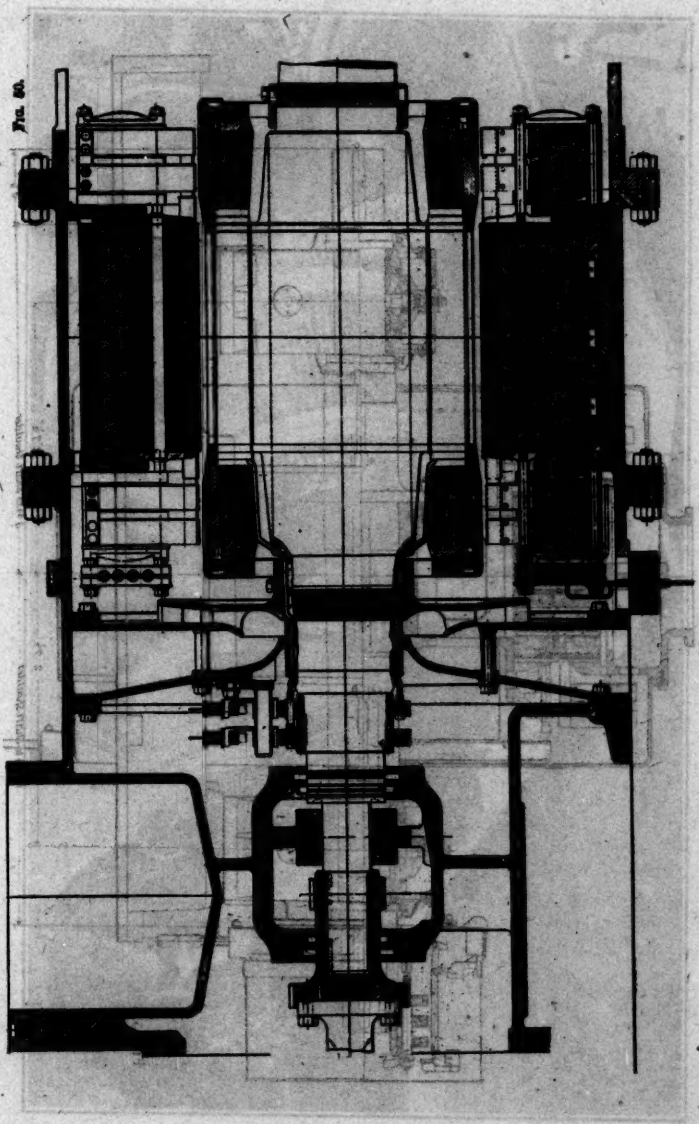


FIG. 60.

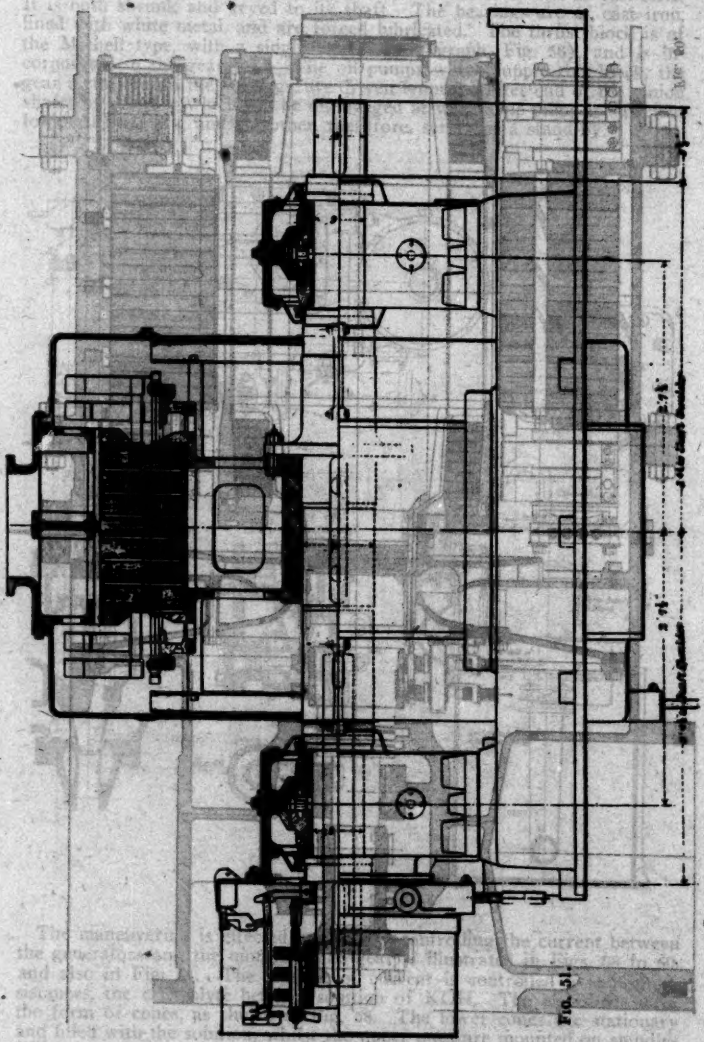


FIG. 51.

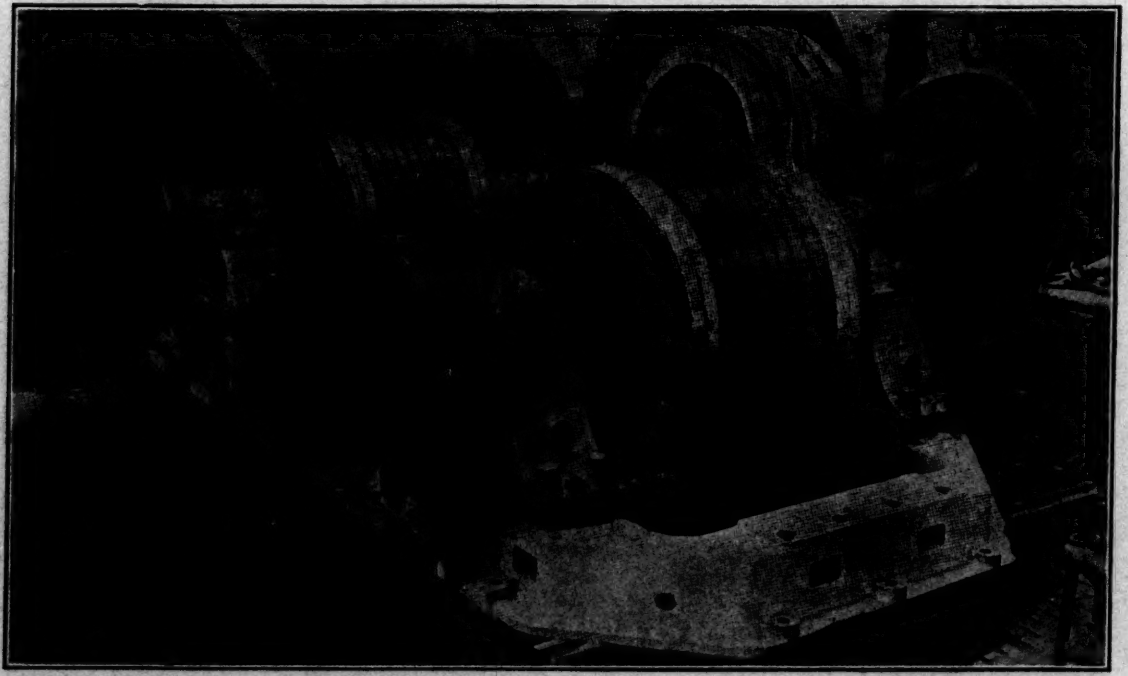


FIG. 52.—ROTORS FOR INDUCTION MOTORS.

FIG. 53.—INDUCTION MOTORS IN COURSE OF ERECTION.

FIG. 53.—INDUCTION MOTORS IN COURSE OF ERECTION.

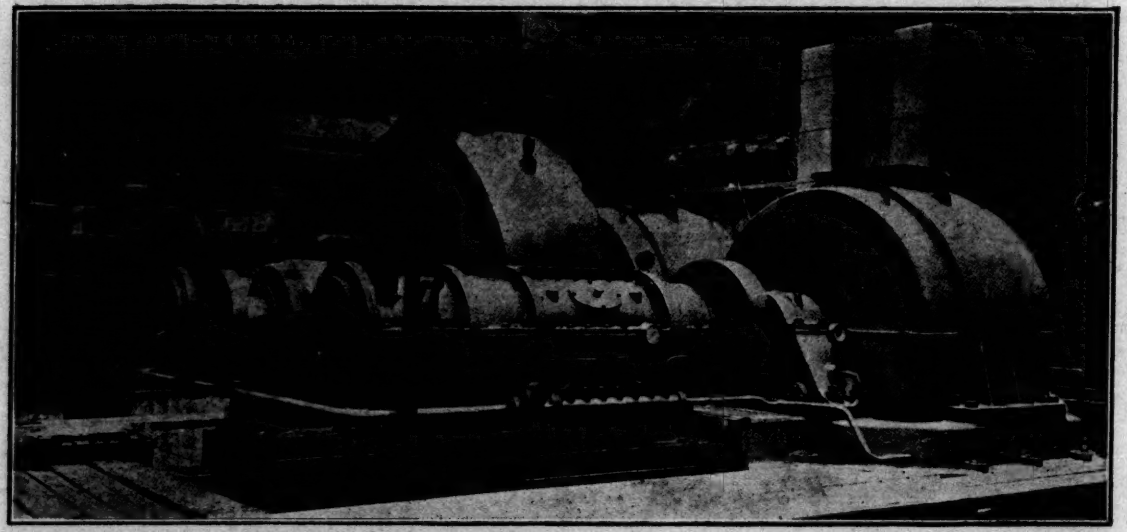


FIG. 55.—MOTORS, CONTROL GEAR AND REDUCTION GEAR ERECTED IN SHOPS.

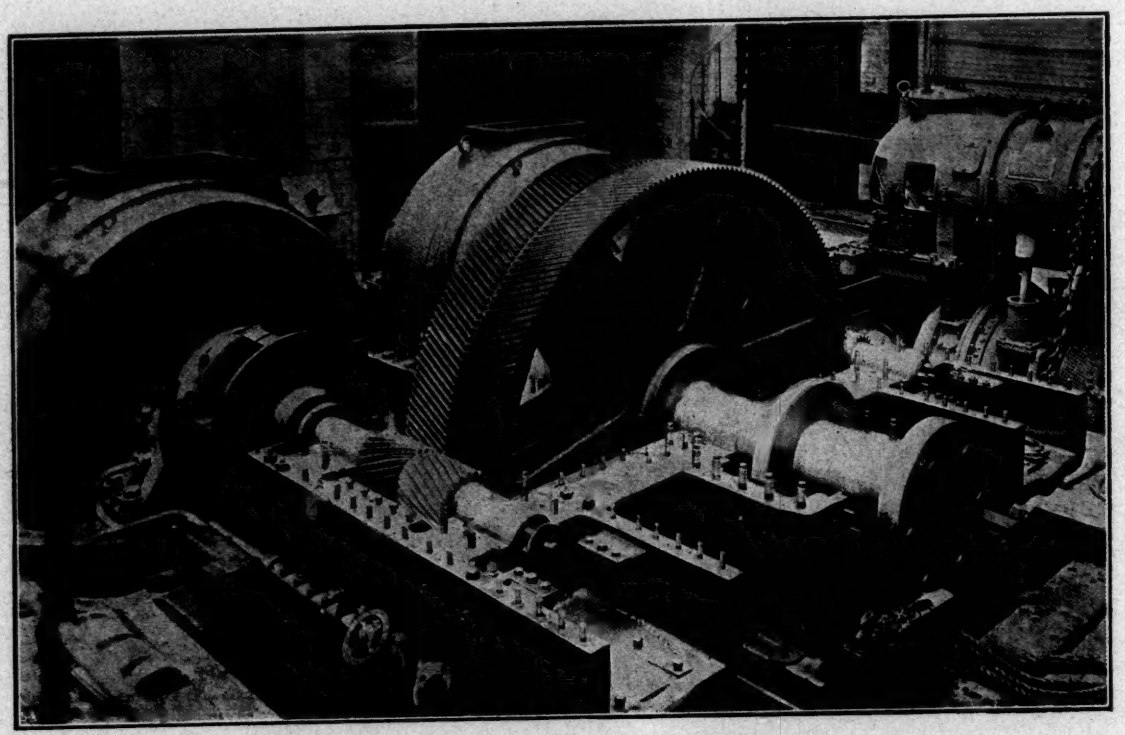


FIG. 56.—GENERAL VIEW OF DOUBLE HELICAL REDUCTION GEARING.

FIG. 56.—GENERAL VIEW OF DOUBLE HELICAL REDUCTION GEARING.

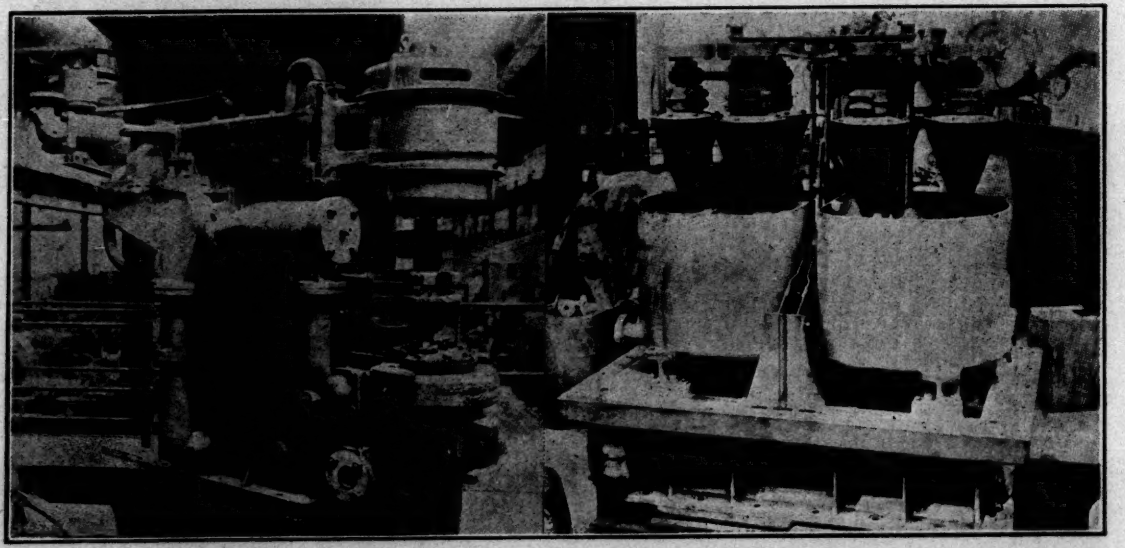


FIG. 57.—PUMPING PLANT FOR CONDENSERS.

FIG. 58.—VARIABLE RESISTANCE FOR CONTROLLER.

FIG. 59.—FRONT VIEW OF CONTROLLER.

FIG. 60.—SIDE VIEW OF CONTROLLER.

If the hand wheel is moved from the mid or stop position in any direction, the propeller turns in a similar direction, its speed correspondnig to the angle through which the hand wheel is turned. At full speed, therefore, either for ahead or astern working, the moving cones are in their lowest position. Reversal of the propeller shaft is also obtained from the motion of the hand wheel. This is effected when it is in the stop position, so that the rotors are open-circuited, a change-over switch being operated, which reverses two of the phases in the stator circuit. The cones with their at-

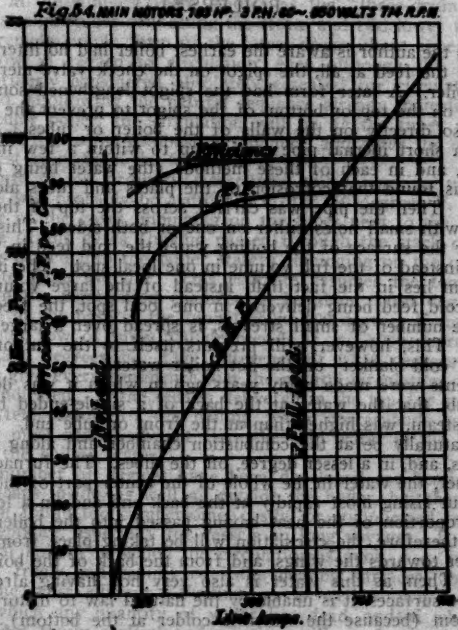


FIG. 59.—FRONT VIEW OF CONTROLLER.

FIG. 60.—SIDE VIEW OF CONTROLLER.

tendant fittings are balanced, so that the power required to operate the hand wheel is only nominal. A stop gear is provided for locking the wheel in any required position. The potash solution is circulated by the pumps shown at the bottom left-hand corner, Fig. 59. A cooler for the electrolyte is arranged at the base of the maneuvering column. This cooler is supplied with sea water taken from the circulating pump discharge.

The various instruments provided are mounted as shown in Figs. 58 and 60. In addition to the usual volmeters and ammeters with shunt regulators, integrating watt-meters are provided, which enable the engineer in charge to obtain in an instant the exact power his machinery is developing. On the top of the board is mounted a synchronizing gear for paralleling the two turbo-alternators, and also an overload cut-out. At full power, the liquid resistance is entirely short-circuited, the current flowing wholly

through a metallic contact. The apparatus permits of the speed being varied with the greatest nicety. It can be reduced to as little as one turn per minute, and at ten turns per minute quite steady running is secured.—"Engineering."

FEEDING AND CIRCULATING THE WATER IN STEAM BOILERS.*

As far as the author is aware the earliest boiler had no internal pipe for introducing the feed at all, the spigot on the check valve merely entering into the boiler. A later form had the spigot lengthened somewhat, and opened out on the top or bottom of the spigot to prevent the water from impinging so directly on the walls of the boiler or tubes. A still later form had a short inernal pipe carried up to within a few inches of the water level, and in each of these methods (the water being delivered in bulk) it was found that corrosion of the plates and tubes alongside was very rapid. Then the pipe was carried across the top of the tubes and perforated with small holes either on one or both sides. This, of course, spread over the surface of the boiling water the cold feed in the form of a "spray," instead of the full volume in one local spot. The improvement of this form lies in the fact that, instead of the large volume of comparatively cold feed being delivered in one local spot, the feed, separated into a large number of small streams, is spread over a large surface of the water. This, however, still presents itself to the author's mind as forming a "cold blanket" on the hot boiler water.

Experiments were made many years ago in which it was demonstrated beyond doubt that the water at the back of a single-ended boiler, when under full steam, was higher than at the front or tube end. The fiercest heat will naturally be at the combustion chamber and along the tops of the furnaces, and, in a lesser degree, off the tubes. The furnaces being so close together, the water in the whole of the center part of the boiler will be rising, but rising most rapidly at the back end and, in a lesser degree (a large proportion of the heat having passed into the boiler water) at the front; therefore, the circulation will be taking place from the center of the boiler towards the wings, and from the back of the boiler towards the front. Then, as this water is also very hot, having already passed over the hot surfaces, it is unable by the natural law to disturb the water at the bottom (because the water is colder at the bottom), but simply flows over and by-passes this colder water and rises again to the surface.

It would appear that there will probably be a difference of about 1,700 degrees F. between the temperature of the gases in the furnace and on combustion chamber plates and the temperature of the gases at the front end of the boiler where the gases pass into the uptake. This figure of 1,700 degrees is estimated, assuming the temperature in the furnace and combustion chamber at about 2,500 degrees F., 500 degrees F. at base of funnel and 800 degrees F. at the outlet of tubes (front end of boiler opening to uptake)—*i. e.*, 2,500 degrees—800 degrees = 1,700 degrees F. These figures, while only approximate, should be sufficient to indicate that the flow of circulation will probably be as indicated. The feed water should be introduced in some way following the lines of flow and facili-

* Abstract of paper read before the Institute of Marine Engineers by Mr. John Watson, on Tuesday, October 8.

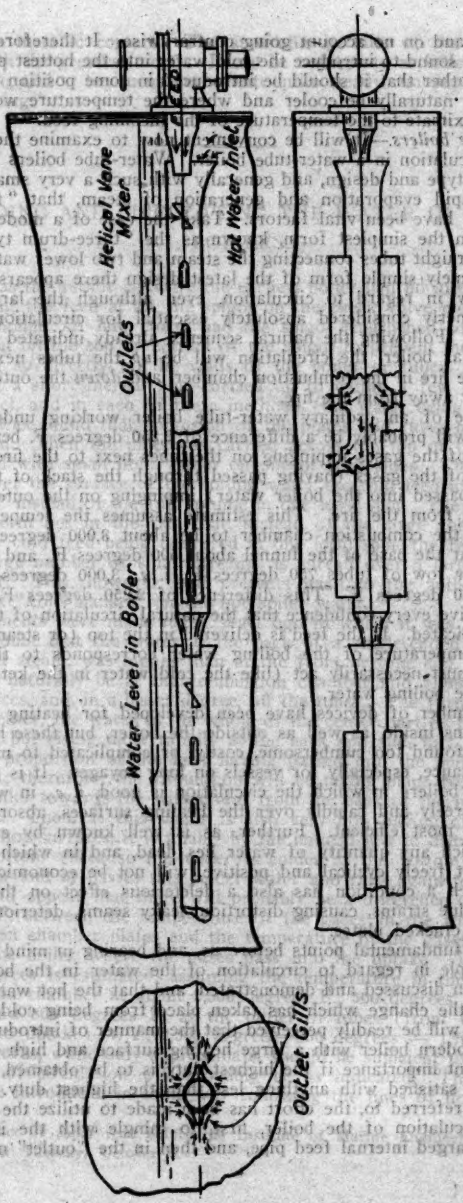
tating them and on no account going contrarywise. It therefore does not appear to be sound to introduce the cold water into the hottest part of the boiler, but rather that it should be introduced in some position where the water would naturally be cooler and where the temperature would more nearly approximate to the temperature of the incoming feed.

Water-tube boilers.—It will be convenient now to examine the probable flow and circulation in a water-tube boiler. Water-tube boilers have been so varied in type and design, and generally with such a very small volume of water, rapid evaporation and generation of steam, that "feed" and "circulation" have been vital factors. Take the case of a modern water-tube boiler in the simplest form, known as the "three-drum type," with practically straight tubes connecting the steam and two lower water drums. In the extremely simple form of the latest design there appears to be no question now in regard to circulation, even although the large down-comers—formerly considered absolutely essential for circulation—are no longer fitted. Following the natural sequence already indicated in regard to the circular boiler, the circulation will be *up* the tubes next to and nearest to the fire in the combustion chamber, and *down* the outer row or rows farthest away from the fire.

In the case of an ordinary water-tube boiler working under forced draft, there will probably be a difference of 2,250 degrees F. between the temperature of the gases impinging on the tubes next to the fire and the temperature of the gases (having passed through the stack of tubes, the heat having passed into the boiler water) impinging on the outer row of tubes remote from the fire. This estimate assumes the temperature of the gases in the combustion chamber to be about 3,000 degrees F., the temperature at the base of the funnel about 500 degrees F., and just outside the outer row of tubes 750 degrees F., *i. e.*, 3,000 degrees—750 degrees = 2,250 degrees F. This difference of 2,250 degrees F. appears sufficient to give every confidence that the natural circulation of the boiler will be as indicated. If the feed is delivered in the top (or steam drum), where the temperature of the boiling water corresponds to the steam pressure, it must necessarily act (like the cold water in the kettle) as a blanket on the boiling water.

A great number of devices have been developed for heating the feed water by means inside, as well as outside the boiler, but these have frequently been found too cumbersome, costly, or complicated to meet with general acceptance, especially for vessels on long voyages. It is generally accepted that boilers in which the circulation is good, *i. e.*, in which the water flows freely and rapidly over the heating surfaces, absorbing the heat, are the most efficient. Further, as is well known by engineers, boilers in which any quantity of water lies dead, and in which the circulation is not freely cyclical and positive, will not be economically efficient, and such a condition has also a deleterious effect on the boiler, producing undue strains, causing distortion, leaky seams, deterioration of material, and cracked plates.

With these fundamental points before us, and bearing in mind that the general principle in regard to circulation of the water in the boiler has frequently been discussed and demonstrated, and that the hot water must, by reason of the change which has taken place from being cold, rise to the surface, it will be readily perceived that the manner of introducing the feed into a modern boiler with a large heating surface and high pressure is of paramount importance if the highest duty is to be obtained, and we should not be satisfied with anything less than the highest duty. In the special device referred to, the effort has been made to utilize the streams of natural circulation of the boiler, first, to mingle with the incoming feed in an enlarged internal feed pipe, and then in the "outlet" nozzle to



AN IMPROVED TYPE OF FEED INLET.

again aid the circulating streams by inducing them through the "gills," so that the feed water will be thoroughly mixed with the boiler water before it actually touches on any part of the boiler elements.

The illustration shows the feed-inlet connection of a boiler, and the inlet for the hot boiler water is shown together with the feed nozzle, which is designed so as to draw in the hot water with the feed, and the helical or suitably shaped vane gives a turning mixing effect to the now united mass of flowing water with a view to thorough mixing. After passing this vane the first nozzle comes into action and a certain portion of the mixed water is delivered into the boiler water. The remaining portion of the water then passes through another simple injector (similarly designed to that where the feed enters the internal pipe), and the same process is repeated through a series of injectors and nozzles, as shown. It will thus be seen that along the whole length of the boiler there will be a continual stream of water—representing the "feed," plus, say, twice its volume—being directed toward the tubes in the hottest part of the boiler. The outlet nozzles are specially designed in series to give an easy flow of the water through them and to draw the hot water through the "gills," so as to further mix and heat the feed.

It does not appear as if there could be any doubt in general principle that the correct place to deliver the "feed" is where the water will naturally be cooler, and where the fresh supply of water can be delivered at as nearly as possible the same temperature as the surrounding water, and this is one of the principal features of the arrangements under consideration. The general principle of the apparatus is thus that the hottest water is taken from the upper surface, and delivered in the more or less "dead" water at the bottom, so that it flows up between the furnaces into the upward hot streams rising from the furnaces and tubes.

The idea of delivering the feed into this "dead" water has been frequently advocated, and in some patented arrangements has been done, but the author has every confidence that the simple apparatus now being described will make its adoption and utility more generally possible and desirable. It appears obvious that heat will pass more readily into the feed water by direct contact with hot water than by injection of steam, by reason of their being more readily mixed. We find that 1 pound of feed water being delivered into the internal feed pipe at 120 degrees, and mixed with only ¼ pound (an addition of 25 per cent) of the hottest water in the boiler, at, say, 387 degrees, would produce water of a mean temperature of 173.6 degrees to be delivered into the boiler; while, if it had been mixed with an equal quantity of hot water in the enlarged internal pipe (an addition of 100 per cent hot water instead of 25 per cent), the mean temperature (if thoroughly mixed) would be 237 degrees. With "feed" water being delivered at the boiler at 200 degrees, and being mixed with twice the quantity of the hottest water in the boiler (corresponding to 200 pounds per square inch), the water as actually delivered into the boiler would be about 333 degrees; and this we estimate could be attained for a feed-supply pipe with check valve 2½ inches bore by an internal pipe enlarged to only about 4½ inches diameter, with fittings, as shown. It will be observed here that with feed at 120 degrees into a boiler working at 100 pounds pressure per square inch, the advantage gained by mixing the feed inlet, as proposed, with twice the quantity of the hot boiler water, would be to raise the temperature of the water delivered at the outlet nozzles by 180 degrees, while with feed at 200 degrees the outlet would be raised by 68 degrees. From this will be evident the advantage that would be gained in installations where no external feed heater is used. With surface feed heaters there is, of

course, heat loss in passing through the tubes and frictional loss in driving the water through them, producing additional "head" on the pumps, and it has generally been found impracticable to heat the feed to more than 200 degrees F. at the pumps, while this temperature in long lengths of feed pipes will be considerably reduced before reaching the boilers—probably down to 190 degrees or 185 degrees—so that, even where these are fitted, there appears ample scope for the devices under consideration to be of very considerable advantage in safe and economical feeding.

In vessels with a large number of auxiliaries and in turbine vessels it may probably be desirable to use a feed heater for utilizing any surplus exhaust steam not required by turbines or evaporators, with the fitting described, to complete the heating of the feed and promote circulation, but in many vessels the feed can be effectively dealt with in the boiler itself by the means which we have endeavored to describe, without any further encumbrances in the way of external feed heaters.

It will readily be seen that while it may be necessary to obtain a high degree of economy throughout by utilizing exhaust steam for feed heating purposes, *i. e.*, condensing this steam by imparting its heat in this process to the feed, yet, even under these conditions, it is often necessary to put a considerable quantity of cold water into a boiler. The "circo-feed" heater under these conditions cannot fail to be of distinct value in reducing the bad effect of the cold water on the boiler. For if, as already indicated, it be arranged so that the incoming feed be mixed with an equal quantity of the hot water in the boiler, the cold water would be heated to approximately double the temperature before delivery into the boiler, and it is manifest that the circo-heater has only to be enlarged at a small extra cost, so that the feed mixes with twice the quantity or more, that a greater benefit may be derived.

We therefore consider that whether a feed heater be adopted for heating the feed for economical purposes or not, the fitting of a circo-heater ensures that whatever the temperature of the feed, its temperature is raised considerably (from, say, 25 per cent to 150 per cent, according to the size of internal pipe and nozzle, temperature of the feed, and the temperature corresponding to the pressure of the boiler) before the feed water is actually mingled with the body of the water in the boiler. There are no valves, springs or other working fittings to be adjusted or to get out of order, and that if properly designed and fitted with suitable materials it should not require any attention, except an occasional overhaul to ensure that the pipes and nozzles are clear.

It will be perceived that there can be no direct economy, for the simple reason that the fitting is self-contained in the boiler, and consequently no heat is saved which would otherwise be lost.

But while this is true, we may fairly expect an indirect economy, due to steadier steaming under all conditions—a direct economy in repairs and, in addition, lengthening the life of the boiler, with a greater confidence in its safety. These are surely valuable assets to the owner and his clients, as well as to the sea-going engineer. It is, generally speaking, only owners and superintendents who have the opportunity of fitting and testing proposals of this nature, and the desire to test depends upon how sound the proposals are, and whether they appear to offer something which, if attained, is worth having and will provide an adequate return for the outlay. How far these conditions have been fulfilled remains to be seen, but the effort has been made to attain an efficient natural circulation of the water in the boiler and at the same time heating the cold feed before delivering into the boiler in the simplest possible manner.—"Shipbuilding and Shipping Record."

WATER GAGES ON MARINE BOILERS.

(Abstract.)

By W. M. McRobert.

Notwithstanding that most examiners of applicants for marine engineers' licenses make a special feature of the water gage, numerous boiler accidents and explosions occur either from engineers not being familiar with the construction of the water gage or not satisfying themselves by actual test that the cocks, pipes and connections to the boiler are clear.

The writer recalls one instance in his experience where a "donkey" (an auxiliary) boiler was practically ruined owing to a misunderstanding among engineers, who may be called, for convenience, Nos. 1, 2 and 3. It was the night before leaving port, and steam was being raised on the main boilers preparatory to proceeding on a voyage to Australia. The auxiliaries, including cranes, light engines, pumps, etc., were being supplied with steam from the donkey boiler, which would be in use only for a short while before changing over to the main units. Engineer No. 1 was about to go off watch and before doing so "pumped up" the donkey boiler; but during the time this operation was in progress he had other duties to attend to, with the result that he allowed too much water to get in the boiler. The gage showed more than a full glass.

The feed pump which was used while in port to supply the donkey boiler with water was not required when at sea, and engineer No. 1, knowing that he had put too much in the boiler, came to the conclusion that the pump would not be further required and closed the cock on top of the boiler which controlled the steam supply to the pump. This cock was located near cock *A*, Fig. 3. Engineer No. 2 now came on watch, and No. 1 mentioned to him that he had pumped "a little too much" water into the boiler, but added that it would soon be all right. After being an hour on watch No. 2 still found the glass full and frequently blowing through by means of cock *E*, Fig. 3; water still issued forth, and he naturally expected that No. 1 must have pumped a lot more water into the boiler than he reported. However, No. 2 thought he would "wait a bit;" but after three hours the glass still showed full, and he then decided to blow water out of the boiler until the water level in the glass could be determined, but even after doing so the gage still showed full.

Engineer No. 3 next came on watch and after thoroughly blowing the glass by means of cock *E*, Fig. 3, and learning the account of No. 2, he began to investigate and on going on top of the boiler found that cock *A* was shut. On opening it the water in the glass disappeared. He then thoroughly tested the gage by means of the cock *T*, and on opening it only dry steam issued. It was found that Engineer No. 1, when shutting off the steam supply to the feed pump as he supposed, had by mistake closed cock *A* which controlled the pump steam line.

Many steamers have had similar experiences through false water level, and on examination it has been proved on nearly every occasion that the cocks or connections to the steam and water spaces were closed or choked. This cannot really be attributed to accident, but to lack of care or to bad management.

Unless an engineer is able to prove that he is thoroughly acquainted with the water gage and able to test its veracity in every way, he should on no account be left in charge of a marine or any other type of boiler. The writer does not favor globe valves on water-gage connections; straightway cocks are most reliable.

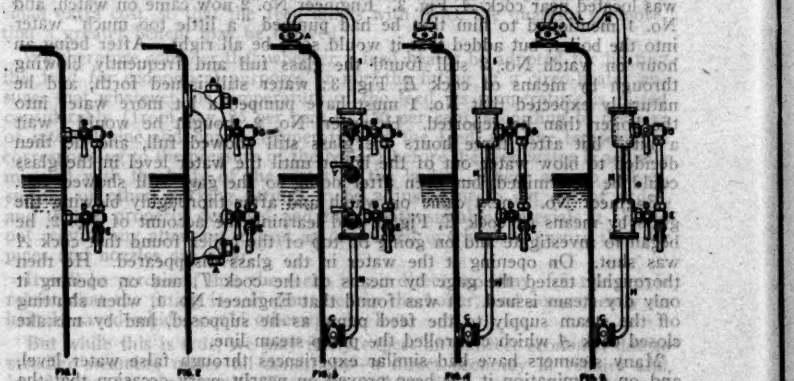
Figs. 1, 2, 3, 4 and 5 serve to illustrate the most usual methods of attaching gage mountings to marine boilers. Figs. 1 and 2 are not to be recommended, although on the old type of boilers they are sometimes found.

The principal features of each gage and the method of proving the correctness of the water level are separately dealt with in the following.

Referring to Fig. 1: It will be seen that the gages are screwed into the boiler front head. They may be tested as follows: First let *B* remain open, then close *D* and open *E*, and if steam issues it shows that *B* and the passage leading to it are clear. Should no steam or water issue, either cock *B* or the passage leading to it is choked and must be cleared. Second, close *B* and open *D* and *E*, and if water issues *D* and the passage leading to it are clear.

Referring to Fig. 2: In this case the gage cocks are attached to a bent pipe of about 3 inches internal diameter, the top end communicating with the steam space and the lower with the water space of the boiler. The large pipe is not likely to become choked and is superior to the design shown in Fig. 1. The gage can be tested in practically the same manner as Fig. 1. Removable plugs are fitted at *AA* and *BB*, by which the openings may be cleared when necessary by means of a rod or wire, when the boiler is not under steam.

Referring to Fig. 3: In this design there is a clear opening from *A* to *C* through the column *Y*, and to "blow through the glass" it is only necessary to shut cocks *D* and *B* alternately, keeping *E* open. But to "blow through the water gage," including the pipes *H* and *I*, it is necessary, after blowing through the glass as described, to shut *A* and *C* alternately, at the same time keeping *B*, *D* and *E* open for sufficient time to completely discharge the contents of the gage and its connections.



FIGS. 1 TO 5 TYPES OF WATER GAGES FOR MARINE BOILERS SHOWING CONNECTIONS

When *B*, *D* and *C* are clear and *A* choked, the steam remaining in the glass and in the pipe *I* leading from column *Y* to *A* becomes condensed and the water flowing through *C* to take its place rises in column *Y*, also in the glass to a level above that of the water in the boiler. In other words, the gage shows a false water level.

If cock *E* be now opened and water is blown out, then on *E* being again closed the water in the gage will rise higher than before, and be still further misleading. On the other hand, when *B*, *D* and *A* are clear and *C* choked, the water, if any, in the glass is trapped and no longer rises and

falls with the water in the boiler or with the motion of the vessel; it, however, slowly rises in the glass owing to the condensation of the steam in the upper part of the gage, until such time as *E* is opened, when all the water in the glass is blown out, and on *E* being again closed the glass will not show any water, notwithstanding that the water in the boiler may be at the proper level.

When test cocks *T T T* are attached to column *Y*, as shown in Fig. 3, they cease to be reliable when either *A* or *C* or the pipe in connection therewith is choked wholly or nearly so.

Referring to Fig. 4: Sometimes the water-gage fittings are arranged as shown in Figs. 4 and 5 with no passage up the column, the central portion *N* of the column being simply a pillar or connecting piece of any convenient section between the upper and lower portions to which the cocks *B* and *D* are attached. By this arrangement double communications are eliminated and there is no need for what is known as the "double shut-off" in testing the accuracy of the gage. When, however, the gages are constructed in this manner, the cocks *B* and *D* are unreliable as test cocks in the event of there being no glass in the gage. This feature should be carefully noted. Moreover, when in working condition the reduction of pressure in the glass which arises when *E* is opened causes the water in pipe *H* to rise above its normal level. This objectional feature should also be noted.

Referring to Fig. 5: Sometimes due to faulty construction, there may be a bend *L* in the steam pipe *I* leading from cocks *A* to *B*. This has occurred when new boilers have been installed on ships, the bend being caused in an effort to clear other pipes, beams, fittings, etc., located near the smokebox. With such a bend the condensed steam collects in the pipe and falls to the bottom of the bend, and in time completely fills the pipe *J* to *K*. The steam from *K* to the level of the water in the glass is throttled and, as condensation proceeds, leads to a reduction of pressure in the pipe below that of the boiler, with an equivalent rise of the water in the bend and also in the gage glass. When the ship is not in motion, the water in the gage glass increases in height until cock *E* is opened, or until the pressure in the boiler is so much in excess of that in the lower part of pipe *I* as to cause the water in the bend to be blown into the gage glass. In either case immediate change of water level takes place.

In the ordinary course of working, the condition described is modified by the presence of air in the upper part of the gage and by the rise and fall of the water in the boiler and gage glass arising from the motion of the steamer.

Probably more than half of the steamers afloat are fitted with water gages as shown in Figs. 3 and 4, and it is important that engineers should be especially acquainted with their construction and principle of operation. Cocks *A* and *C* are not necessary for the testing of gages as shown in Figs. 4 and 5. Engineers should note that it is impossible to reliably test water gages as shown in Fig. 3 when the cocks *A* and *C* are absent. They should note the effect which the choking of cock *A* or *C* or pipe *H* or *I* has on the indications of the test cocks *T T T* fitted on column *Y*.

When a gage glass is too short or is placed either too high or too low in the fittings, it is liable to become choked by the packing being forced over its ends when the gland is being screwed up.

When a gage glass is too long it will not permit of a rod being inserted at *G* to allow for cleaning the passage through cock to boiler. This defect would also tend to cause a false water level.

Boiler accidents have occurred from the handles of the cocks not being exactly at right angles to the cock openings, owing to the former being strained and becoming bent or twisted.

Another casualty took place due to the twisting of the stem on cock *A*, Fig. 3, from its original position, resulting in the cock being shut when apparently open. Avoid straining the stem or handle of the cock.—"Power."

2500-H.P. RATEAU MARINE GEARED TURBINES.

It is certain that the steam turbine will continue to play a very important part in connection with marine propulsion, and it is equally certain that the direct turbine drive cannot perfectly satisfy all the requirements of marine engineers. There remain, therefore, at present only two ways of utilizing the steam turbine satisfactorily for ship propulsion, the first being in connection with an electric generator and motor, and the second being through the intervention of speed reducing devices. With single reduction gearing, the highest degree of economy is unattainable in mercantile vessels with the usual propeller speeds, because in order to keep the gear-wheel diameter down to reasonable dimensions, the turbine must be operated at comparatively low speeds. The double reduction gear, although doubtless not quite so mechanically efficient as the single reduction, enables the turbines to be run at the highest practical speeds, thereby reducing the size of the turbine. The gearing can, of course, be designed so that the power for each turbine is transmitted to the final wheel independently, giving an even distribution of pressure on the teeth. With two turbines and two separate high-speed gear trains, greater reliability is secured, for even if one turbine is entirely out of commission, the voyage of the vessel can be continued.

We have had an opportunity of inspecting in operation a 2,500 shaft horsepower Rateau turbine set and double reduction gearing at the works of the British Westinghouse Company, Limited, Tracord Park. The set is of a standard type which this firm is now building, and is intended for a ship ordered by the Moss Steamship Company, Limited, Liverpool, from the North of Ireland Shipbuilding Company, Londonderry. The dimensions of the vessels are as follows: Length, 385 feet 6 inches; length between perpendiculars, 370 feet; breadth, extreme, 51 feet 3 inches; breadth molded, 51 feet; depth molded to upper deck, 28 feet 2 inches; depth molded to bridge deck, 36 feet 2 inches; deadweight, 7,200 tons; draught, 23 feet 11 inches; and speed, 11 knots. The normal shaft horsepower required is 2,500, the speed of the propeller 70 revolutions per minute, and the turbine revolutions 3,000 per minute. The astern power required is 1,800 shaft horsepower.

The turbines are of the Rateau impulse type, a pattern which has been brought to a high pitch of perfection for land purposes, and which adapts itself admirably to marine propulsion. The general arrangement of the unit comprising turbines, gear and condenser, is shown diagrammatically in Fig. 1, both the high and low-pressure turbines as erected in the makers' works, and Figs. 2 and 3 show the general arrangement of the turbine gear. It will be observed that the turbine consists of one high and one low-pressure cylinder mounted side by side, both driving through high-speed pinions the double reduction gearing, which in turn drives the propeller shaft. The high and low-pressure cylinders are each divided into two sections, one for steaming ahead and the other for going astern; as more clearly shown in the diagrammatic views, Figs. 4 and 5. For ahead steaming, steam is first expanded in the ahead nozzles A, performing work in the velocity-compounded stage B, after which it passes through several single-impulse stages C. The steam is then led across through interconnecting piping to the ahead nozzles of the low-pressure cylinder D, and passes through the low-pressure single-impulse stages to the condenser outlet E.

For astern steaming, the steam after performing work on the astern velocity-compounded stage F, finally passes through the astern low-pressure stages G to the condenser, which is placed immediately below the low-pressure half of the machine.

Being of the impulse type the pressure and temperature inside the high-pressure casing is kept low, as a much larger expansion is allowed in the first than in the succeeding stages, while in the low-pressure cylinder the pressure need only be a few pounds above that of the atmosphere. The aft ends of the cylinders are bolted direct to the gear case, and the forward

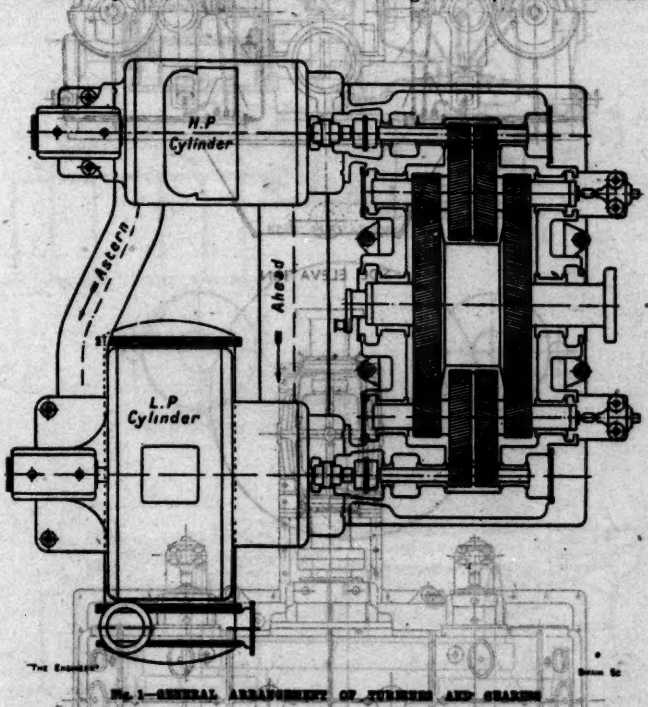


Fig. 1—GENERAL ARRANGEMENT OF TURBINE AND GEARING

ends have sliding seatings to allow of expansion due to temperature changes. The cylinders and diaphragms are divided horizontally to facilitate opening up. The nozzles on the ahead section of the turbine are usually divided into three groups, each controlled by a separate valve. Each group consists of a different number of nozzles, so that by admitting steam to the groups separately or to various combinations of groups, throttling losses can be avoided at seven different ahead speeds. The nozzles for the astern section of the turbine comprise one group only, from which a power of usually 70 per cent of the normal ahead power can be obtained for astern steaming. The complete nozzle block containing a number of nozzles is built up of two forged-steel angles, machined all over, between

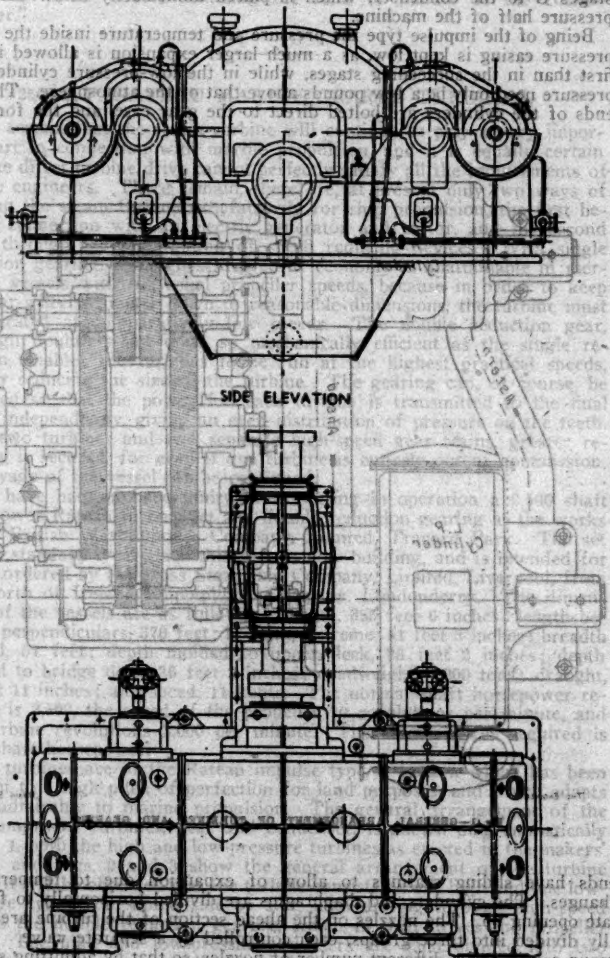


FIG. 2.

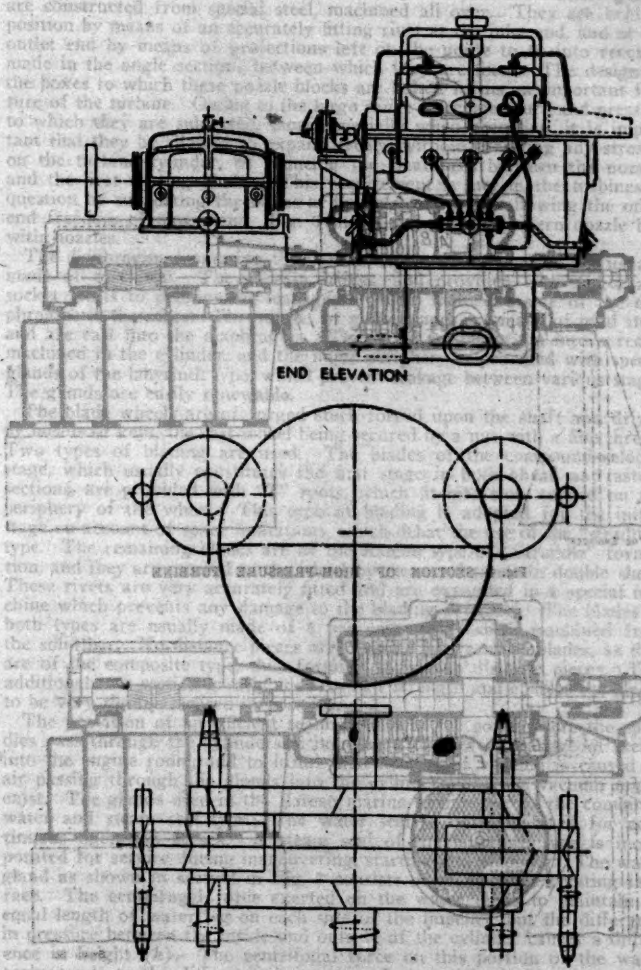


FIG. 3.

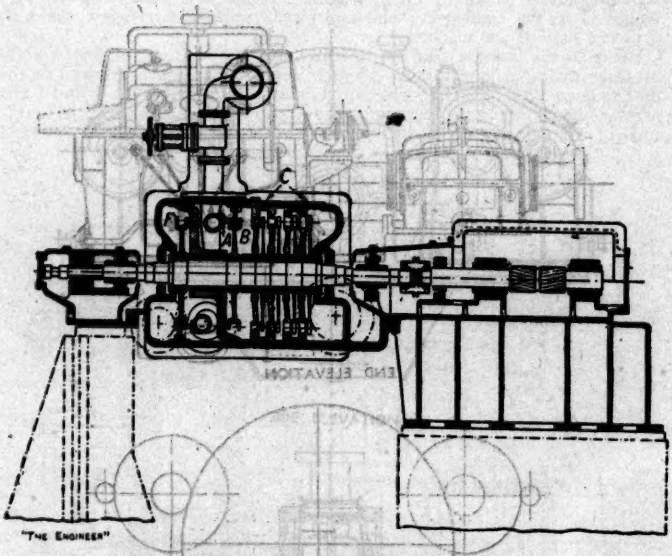


FIG. 4.—SECTION OF HIGH-PRESSURE TURBINE

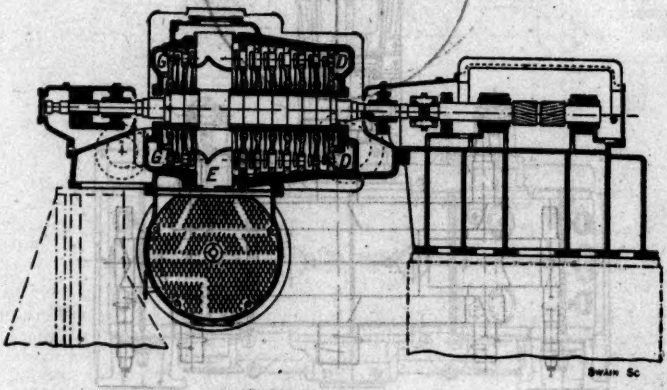


FIG. 5.—SECTION OF LOW-PRESSURE TURBINE

which the guide plates which form the nozzles are inserted. These guides are constructed from special steel, machined all over. They are held in position by means of an accurately fitting rivet at the inlet end, and at the outlet end by means of projections left on the guide to fit into recesses made in the angle sections between which they are fixed. The design of the boxes to which these nozzle blocks are bolted forms an important feature of the turbine. Owing to the large range of temperature and pressure to which they are subjected, more especially when reversing, it is important that they be allowed to expand freely, without throwing any stresses on the turbine cylinder, or reducing the clearances between the nozzles and the first moving blades. This requirement is met in the turbines in question by supporting the boxes at the inlet end only, allowing the other end freedom of movement. Fig. 6 shows the complete astern nozzle box with nozzles.

The diaphragms separating the various stages of the turbine are usually made of cast iron. They are in halves, and provided with spigot and socket joints to prevent the leakage of steam from one side of the diaphragms to the other. The nozzles or guide blades are made of mild steel, and are cast into the diaphragms. The outer periphery fits into a recess machined in the cylinder, and the inner periphery is provided with special glands of the labyrinth type, which prevent leakage between various stages. The glands are easily renewable.

The blade wheels are of forged steel, forced upon the shaft and driven by means of keys, the cast wheel being secured by a nut with a fine thread. Two types of blading are used. The blades of the compound-velocity stage, which usually constitutes the first stage in both ahead and astern sections, are provided with "T" roots, which fit into slots turned on the periphery of the wheel. This type of blading is adopted for the initial stage on account of space limitations, which debar the use of the "straddle" type. The remaining blades are of the Rateau type of "Straddle" formation, and they are secured to the wheel by means of rivets in double shear. These rivets are very accurately fitted and are expanded in a special machine which prevents any damage to the blading or wheel. The blades of both types are usually made of 5 per cent nickel steel, machined from the solid bar. No distance pieces are required between the blades, as they are of the composite type, thus forming their own distance pieces. This additional root section greatly adds to their strength and enables the blades to be very rigidly secured in position.

The provision of an efficient sealing gland at the point where the spindles pass through the cylinders is important, to prevent leakage of steam into the engine room, and to insure that no loss of vacuum is caused by air passing through the glands into the cylinders where a vacuum might exist. The glands used in the Rateau marine turbine are of the combined water and steam seal type. The water seal is normally used for continuous operation at sea. A steam seal of the labyrinth type is incorporated for service during maneuvering, starting and stopping. The water gland as shown in section in Fig. 7 consists of an impeller rotating in a race. The centrifugal force exerted on the water tends to maintain an equal length of water leg on each side of the impeller, but the difference in pressure between the inside and outside of the cylinder causes a difference in height (h). The centrifugal force on this portion of the water corresponds to the difference in pressure between the inside and outside of the cylinder. The water used for sealing purposes is entirely recovered and no loss of fresh water occurs. With steam sealing, the admission of steam to the gland is regulated so that no loss of steam or condensate occurs, and as in the case of the water seal, the steam or condensate passes to the main condenser. When the amount of steam has once been adjusted no further regulation is necessary, as the conditions in the gland are independent of load or speed.

An overspeed governor, automatically to prevent racing is shown in section in Fig. 8. This device is fitted on the end of the turbine shaft. It consists of an unstable governor in which the weight, having its center of gravity a small distance from the center of rotation, is opposed by a spring of such strength that the force exerted by it is less than the increase in centrifugal force due to the weight moving further from the center of rotation. Since this force is cumulative, immediately the weight begins to move it flies out and trips the emergency valve through the agency of levers shown. Two such governors are provided on each shaft. Relief valves are provided on both high and low-pressure cylinders to relieve any excess of pressure in case of breakdown of the condenser auxiliaries.

The main bearings are separated from the glands by small isolating chambers which separate the cylinder from the bearing housings and casing, and in which the parts requiring lubrication are situated. The covers of the bearing housings can be readily removed without disturbing any other part of the machine. The main bearings are spherical. They are made of cast iron in halves, and lined with white metal. At each side and on top and bottom a steel pad is provided under which there are liners of varying thickness, thus enabling the shaft to be adjusted in any direction for lining up and allowance for wear. A thrust collar is provided on each shaft to maintain correct alignment relative to the cylinder. Lubrication is effected by a combined gravity and pressure system. Two positive rotary pumps direct connected to the intermediate-gear shafting deliver oil to the supply tank, from which oil flows by gravity to the bearings. A valve is also provided to enable the oil to be delivered direct to the bearings if necessary. Oil coolers are supplied in duplicate. The turbine shafts are coupled direct to the gear pinions by flexible couplings.

The gearing is the product of the Power Plant Company, Limited, West Drayton, and is of the double-reduction helical type, giving a reduction of 43 to 1 in two sets of gears, 6 to 1 and 7.1 to 1, respectively. There are two independent high-speed trains driven from the high and low-pressure turbine spindles, working on to a common wheel for driving the propeller. The high-speed pinions are of oil-treated nickel steel with 3½ to 5 per cent nickel, and .25 to .35 per cent carbon. The high-speed wheels are built up from forged-steel rims shrunk on and otherwise securely fastened to cast-iron disc centers. The two low-speed pinions are made from .5 carbon steel, and are solid with their shafts, which carry on a swell portion in the center the high-speed wheels. The low-speed wheel is usually of cast steel with forged-steel rims shrunk on. The main portion of the gear case is one solid casting, with which are cast in one all bearing housings. The casting is made with box shapes right through with flat outer surfaces. This construction insures great rigidity with comparatively low weight. High-speed and intermediate-speed bearing housings lie in the same horizontal plane; the low-speed bearing housings lie in a lower plane. All necessary oil passages for lubrication of bearings and gears are formed in the body of the case, where necessary lined with brass tubes, ending in bosses at the outside of the gear case into which are screwed substantial gun-metal unions. The gear case cover is made in five separate pieces any one of which can be removed without affecting the others. Oil for lubrication is fed to the high-speed trains by means of oil shields, which insure uniform distribution with either direction of rotation. The low-speed trains are lubricated by means of sprayers, which are easily removable. All the oil from the main bearings and gears drains into the bottom of the case, from which it circulates from the oil pump, strainers and coolers back to the supply tank.

The condenser is of the surface type, with 2,118 square feet of cooling surface, and with water flowing through the ¾-inch tubes. It is attached direct to the exhaust flange beneath the low-pressure cylinder. The body

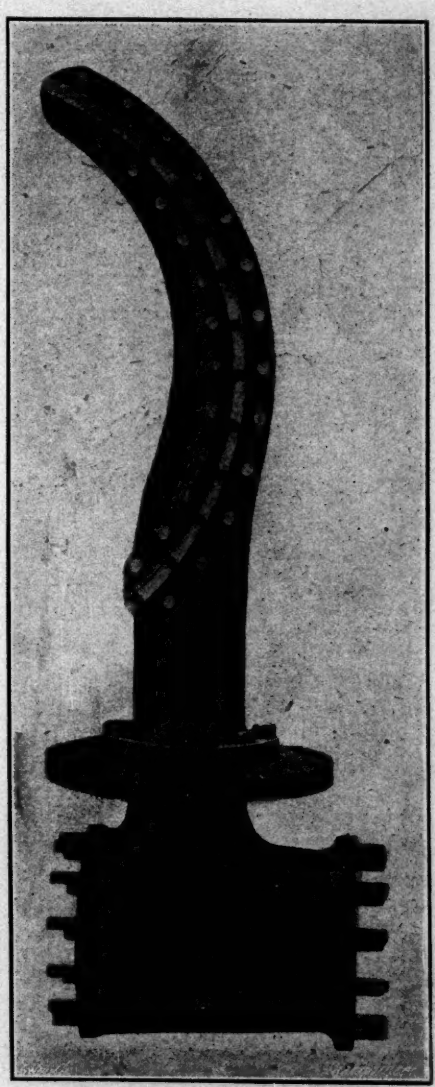
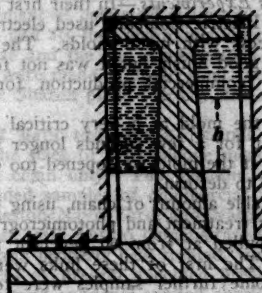


FIG. 6.—ASTERN NOZZLE BLOCK.

is made of cast iron with tubes of solid drawn Admiralty mixture, secured to rolled brass plates by means of screwed ferrules and tape packings. The ferrules are designed to permit freedom for expansion and prevent creeping. The tube plates are stayed to the condenser and corners, the stays being in tension when in service. The circulating pump was made



"The Engineer" Swaim Sc.

Fig. 7.—WATER GLAND

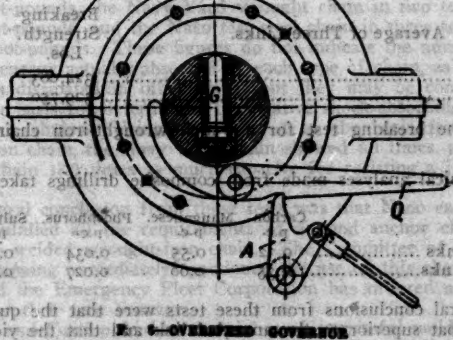


Fig. 8.—OVERSPEED GOVERNOR

by Drysdale and Company, Ltd., and the dual air pump by G. & J. Weir, Ltd. The plant is designed to work with steam at 180 pounds pressure, 100 degrees F. superheat, and 28½-inches vacuum with 30 inches barometer. The total weight of turbines, gears, condenser and pumps is 72 tons.—"The Engineer."

ELECTRIC CAST STEEL ANCHOR CHAIN.*

By H. Jasper Cox.

Wrought-iron chain, either hand or steam-hammer welded, has been used almost exclusively in ships to this day. Mild steel chain has been tried, but has so far proved either too ductile to retain its form under stress or too hard to ensure reliable welds.

General Electric Company's Experiments.—In their first attempt to make cast-steel chain, the General Electric Company used electric-furnace steel, adopting the die-casting process with metal molds. The links produced were perfectly sound and of good finish, but it was not found practicable to develop this process for commercial production for the following reasons:

It was necessary to open the mold at a very critical time, while the metal was congealing; if left for a few seconds longer the links broke, due to the core being solid. If the mold was opened too quickly, the link being in a plastic state tended to deform.

They then made a considerable amount of chain, using sand molds and cores, experimented with heat treatment and photomicrographs, and eventually obtained links which pulled at from 60 per cent to 100 per cent above wrought-iron chain. The first of these links were tested at the Bureau of Standards, and some further samples were tested at a later date at the Charleston Navy Yard. In the latter tests six cast-steel links, 2 inches in diameter, were used; three of the links had been annealed at 850 degrees C. and cooled in the furnace, and three links had been heated to 850 degrees C. and quenched in water. The links were tested in an 800,000-pound chain-testing machine with the average results set forth in the next column.

Before testing, each link was subjected to a proof stress of 145,400 pounds, and none of the links showed a measurable elongation. The diameter was slightly less than 2 inches and the original length 11 13/16 inches.

Average of Three Links.	Breaking Strength. Lbs.	Total Stretch. Ins.
Annealed	324,083	$1\frac{5}{16}$
Quenched	329,750	$1\frac{1}{2}$

NOTE.—The breaking test for a 2-inch wrought-iron chain is 225,792 pounds.

The chemical analyses made from composite drillings taken from the links were:

	Carbon. p.c.	Manganese. p.c.	Phosphorus. p.c.	Sulphur. p.c. ?	Silicon. p.c.
Annealed links	0.32	0.55	0.034	0.025	0.14
Quenched links	0.35	0.68	0.027	0.026	0.15

The general conclusions from these tests were that the quenched link was somewhat superior to the annealed link and that the yield-point of both was much in excess of that of wrought iron. In all these links the stud was cast in.

Following this a series of comparative percussion and bend tests were carried out on separate links of cast steel and wrought iron. The general conclusion from these tests was that they had little relative value, or did not duplicate conditions met with in service.

* Paper read before the American Society for Testing Materials.

National Malleable Castings Company's Experiments.—At this time the subject was brought to the attention of the National Malleable Castings Company, of Cleveland, which immediately saw the resemblance between the problems of cast-steel anchor chain and car-coupler knuckles.

After exhaustive tests and experiments some years ago this company had developed a special electric steel which combined with a high tensile strength a remarkable ability to withstand shock without distortion. This material they called Naco steel. They were invited to participate in the further development of cast-steel chain and prepared samples of 1⅞-inch, 2-inch, 2¼-inch and 2⅞-inch chain, including both couplets of the common link of each size and the end combinations of special links and shackles. These samples were made of Naco steel and were tested on March 1 and 2, 1918, at their Sharon plant. Briefly, the results may be summarized as follows:

A total of 17 static tests and 6 dynamic tests were made. The static tests covered the sizes of chain above mentioned, and were largely upon Naco cast-steel chain only. The dynamic tests covered the 2-inch chain only, and included samples of commercial and wrought-iron chain, chain from the Charleston Navy Yard, and Naco cast-steel chain. The primary purpose of these tests was to demonstrate that the Naco cast steel, when put into chain and properly heat treated, would not only give static tensile results in excess of present specified breaking loads on wrought chain, but would withstand suddenly-applied shock stresses better than wrought-iron chains of corresponding sizes.

Of the four different sizes of Naco chain tested statically, the 1⅞-inch chain showed an average breaking load about 73 per cent above the specified breaking load of wrought chain, the 2-inch chain showed 86 per cent, the 2¼-inch showed 75 per cent, and the 2⅞-inch showed 77 per cent, or an average improvement of about 78 per cent.

In the dynamic tests, consisting of tensile shock tests applied through a Master Car Builders' drop-testing machine to short sample pieces of chain, the 2-inch commercial wrought-iron chain in three tests averaged 142,680 foot-pounds, the Navy Yard wrought chain in two tests averaged 623,200 foot-pounds, and the Naco test steel chain in three tests averaged 1,902,400 foot-pounds. These figures do not indicate the number of foot-pounds of energy actually absorbed by each type of chain, as some of the energy was dissipated in other ways; but they may be considered as a comparative basis for determining the relative shock-absorbing capacity of the different chains. Using the figure 1 as the basis for commercial wrought-iron chain, the Navy Yard chain showed 4.3 times, and the Naco cast-steel chain 13.2 times as much capacity for resisting a jerk shock of this nature.

The general conclusion from these tests was that Naco cast-steel chain not only fulfilled all the requirements of a good anchor chain but was superior to welded wrought-iron chain. The Committee of Lloyd's Register of Shipping immediately sanctioned the use of this chain on classed vessels and the Emergency Fleet Corporation has ordered a considerable quantity for their new tonnage.

Molding and Casting Links.—A number of successful molding and casting methods have been tried, by some of which alternate links are precast and inserted in the molds before casting the connecting links; in others all links may be cast at one time; while other methods again are adaptable to either.

A good deal of controversy has arisen regarding the respective merits of gating on the stud or on the side of the link, but it may be said that good links have been produced by either. The stud being the least important part of the link, any defect caused at the gate is not of the same importance as it would be on the side.

It is considered most desirable that only dry sand molds and cores be used, in order to avoid as far as possible the tendency of piping in the casting, especially in the smaller links.

Heat Treatment.—The Naco cast-steel links were subjected to the same heat treatment as Naco car-coupler knuckles. In this treatment the links were placed on a traveller and slowly passed through a heating furnace in which the desired heat is automatically controlled by special contracting pyrometers. At the discharge end of this furnace the links are dropped into a quenching tank of water and after quenching are slowly passed through a drawing furnace on a second traveller.

For the treatment of complete lengths of 15 fathoms of cable modifications in the arrangements of these furnaces and quenching tanks will, of course, have to be made, but it is understood that this has already been worked out by the National Malleable Castings Company.

Specifications.—Following the first series of tests at Sharon, Lloyd's Register of Shipping prepared specifications of requirements for electric cast-steel anchor chain intended to be used in classed vessels, and these specifications with some slight revision have been submitted to Committee A-1 on steel of the American Society for Testing Materials.

It was not considered desirable to lay down any specific requirements with regard to manufacture, but to commence with it was thought necessary to limit the application to the electric furnace process, to require the use of dry sand molds or cores, and to subject the chain to heat treatment.

With regard to the physical tests, it was decided to take advantage of the higher tensile strength easily obtained in cast steel, and to specify a proof load equal to the present breaking load of wrought-iron chain and breaking load of 40 per cent in excess of the present breaking load. This provided a considerable margin of safety at little or no sacrifice, since owing to the importance of weight as a factor in anchor chain it was not permissible to make a corresponding reduction in the diameter.

Sufficient data were not at hand to specify shock tests for all sizes of cable, nor was it considered necessary in the present state of the development to insist on any particular form of shock test for general application. It was therefore decided to cover this test in a general manner in the specifications, such that the classification societies may specify any particular crucial or commercial test considered necessary to determine the shock-resisting qualities of the material.

In the case of the chain to be made by the National Malleable Castings Company, test pieces from each shot of chain manufactured will for the present be subjected to a Master Car Builders' drop test based on the average results obtained with good wrought-iron chain of similar size.—"Engineering."

THE MANUFACTURE OF CAST STEEL CHAINS.*

By Chester K. Brooks, Assistant to President, National Malleable Castings Company.

Several months ago, when the National Malleable Castings Company was invited to participate in the experimental development of cast steel chain, the first problem which required attention was the working out of suitable foundry methods for producing chain links in sand molds, and uniting them into continuous lengths of chain. A number of entirely successful equipments for making the molds was the result, four of which are shown on page 854.

*Paper read before the American Society of Testing Materials.

Connecting up the links into chain was accomplished in either of two ways, the continuous process, in which the entire chain is poured from one heat, into previously prepared and united molds, by filling one interlinked mold impression after another, and the pre-cast link process, in which half the links comprising a complete length of chain are cast separately, shaken out, cleaned, inspected, and made ready for incorporation into the finished chain. Subsequently, the latter links are placed in mold cavities and coupled up into continuous chain by pouring metal into other molds containing impressions for similar links, interlinked with the pre-cast links. Any of the illustrated methods of making molds is equally applicable to the continuous process or the pre-cast link process.

Another part of the foundry problem was the character of the mold. Recognizing the vital importance of sound castings in a product of this character, we came to the conclusion quite early in our experiments that all of the links should be produced in dry sand molds. The methods of application of gates and risers have also required some study, and we have been able to obtain excellent results both in pouring through the stud and through the side of the link.

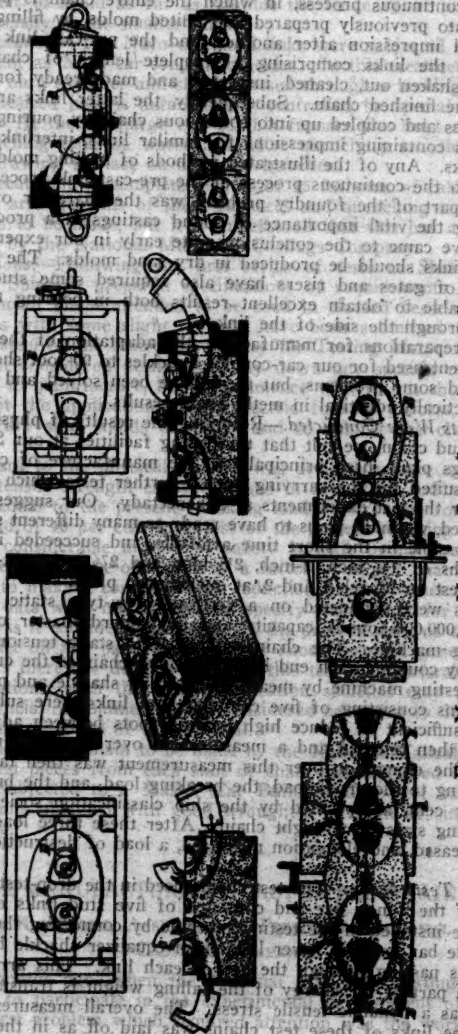
In our preparations for manufacture, the adaptation of the method of heat treatment used for our car-coupler knuckles to 90-foot shots of chain has involved some problems, but these have been solved and the process will be practically identical in method and results.

How Tests Were Conducted.—Referring the results of physical tests on cast-steel stud chain, we felt that the testing facilities at our Sharon, Pa., steel castings plant, used principally in the manufacture of car couplers, were well suited for the carrying out of further tests which seemed desirable after the last experiments at Schenectady. Our suggestion having been accepted, we made plans to have ready as many different sizes of cast chain as possible in the short time available, and succeeded in preparing short lengths of 1⅜-inch, 2-inch, 2¼-inch and 2⅞-inch cast-steel chain, ready for test on March 1 and 2, at our Sharon plant.

The tests were conducted on a vertical screw-type static testing machine of 1,000,000-pounds capacity and a standard master car builders' drop-testing machine. The chains subjected to static tension tests were supported by coupling each end link of the test chain to the cross-head of the static testing machine by means of a testing shackle and pin.

Test chains consisting of five common stud links were subjected to a light load sufficient to reduce high bearing spots between adjacent links, which was then released and a measurement over the three middle links laid off. The elongation over this measurement was then taken at load corresponding to the proof load, the breaking load, and the breaking load plus 40 per cent, as specified by the ship classification societies for the corresponding sizes of wrought chain. After these three loads had been applied, released, and elongation measured, a load of destruction was then applied.

Dynamic Tests Made.—The test chains used in the drop-testing machine were all of the 2-inch size and consisted of five stud links each. These chains were installed in the testing machine by connecting the upper link in a shackle bar and the lower link in an equalizer shackle by means of shackle pins passing through the eye of each link. This arrangement is such that a part of the energy of the falling weight is transmitted to the test chain as a dynamic tensile stress. The overall measurement of the middle three links of these test chains was laid off as in the static tests and the elongation measured after the third blow from each of the specified heights. The weight of the tup was 1,640 pounds, and this weight was allowed to rest upon the equalizer when readings were laid out and elongation measured. In all these tests the numbering of the links was upward, commencing with the bottom link as No. 1.



FOUR METHODS OF MAKING CAST-STEEL ANCHOR CHAIN.

Table I shows the results of all the static tension tests and Table II the results of the dynamic tension tests. Included among the dynamic tests are not only the tests of 2-inch National Malleable Castings Company steel chain, but also certain tests on the same size specimens of commercial wrought-iron chain, and samples from the Boston navy yard. The following details of the tests are of especial interest:

Results of Dynamic Tests.—In dynamic test No. 4, on 2-inch Naco steel chain, the chain was subjected to three blows at 5 feet, three at 10 feet, three at 15 feet, three at 20 feet, 50 at 25 feet and 20 at 30 feet. The

TABLE I.—STATIC TENSION TESTS OF CHAIN.

Test No.	Size.	Weight of Test Chain.	Elongation in. after			Load at Rupture.	Remarks.
			Proof Load.	Breaking Load.	Breaking Load Plus 40 per Cent.		
	In.	lb.				lb.	
1	1½	45	0·03	0·01	0·06	310,000	Broke No. 3 link. Section of metal solid.
2	1½	45	0·06	0·04	0·07	139,700	Broke No. 3 link. Section of metal too small. Slow test.
3	1½	45	0·06	0·04	0·06	152,700	Broke No. 3 link. Section of metal too small.
4	2	120	0·05	0·05	0·13	456,400	Broke No. 4 link. Small thickness on weld. Cracked.
5	2	122	0·08	0·04	0·10	460,000	Broke No. 4 link. Section shows solid.
6	2	122	0·08	0·05	0·08	461,000	Broke No. 4 link. Section shows solid.
7	2	122	0·10	0·07	0·45	391,000	Broke top "B" link. Section shows too small in section.
8	2	122	0·10	0·12	0·03	346,000	Broke bottom "B" link. Section shows too small thickness.
9	2	122	0·07	0·16	Not calculated	310,100	Broke bottom "B" link. Section shows too small thickness.
10	2½	174	0·02	0·00	0·02	456,000	Broke No. 1 link. Section shows solid.
11	2½	174	0·07	0·10	0·05	456,000	Broke No. 3 link. Section shows solid.
12	2½	173	0·06	0·11	0·22	512,500	Broke No. 3 link. Section shows solid.
13	2½	172	0·03	0·05	0·12	505,000	Broke No. 3 link. Section shows solid.
14	2½	171	0·08	0·07	0·10	491,000	Broke No. 3 link. No link examined.
15	2½	[illegible]	0·05	0·03	0·07	483,000	Broke No. 3 link. Section shows solid. See also Dynamic test No. 6.
16	2½	171	0·05	0·09	0·00	764,700	Broke "B" link. Section shows too small in section. See also Dynamic test No. 7.
17	2	[illegible]	0·12	0·19	0·14	467,000	Broke No. 3 link. Section shows small thickness. See also Dynamic test No. 8.

TABLE II.—DYNAMIC TENSION TESTS OF CHAIN.

No. of Test	No. of Links	Material.	Elongation in. after Three Blows at					Total Blows at 25 ft.	Total Blows at 30 ft.	Total Blows at 35 ft.	Chain Broken on	Total Blows to Rupture.	Remarks.
			5 ft.	10 ft.	15 ft.	20 ft.	25 ft.						
		In.											
1	2	Wrought	0·03	0·03	0·13	—	—	—	—	—	10 at 20 ft.	0·15	
2	2	Wrought	0·00	0·03	—	—	—	—	—	—	2nd at 25 ft.	0·03	
3	2	Wrought	0·07	0·03	—	—	—	—	—	—	2nd at 25 ft.	0·03	
4	2	Naco	0·01	0·03	0·03	0·13	0·27	60	20	—	Naco	0·30	
5	2	U.S. Navy	0·10	0·04	0·03	1·27	2·25	6	1	—	6th at 25 ft.	2·25	
6	2	U.S. Navy	0·03	0·03	0·74	1·03	2·05	10	1	—	Link	2·05	
7	2	Naco	0·03	0·03	0·05	0·05	0·10	10	1	—	Naco	0·15	
8	2	Naco	0·00	0·10	0·15	0·10	0·27	10	10	20	10 at 35 ft.	0·38	

[1] Broke No. 2 and 3 links. [2] Broke No. 1 link. [3] Broke No. 1 link. [4] Chain broken in center of link. See also Static test No. 15. [5] Combined fracture in all links. [6] Chain fractured in center of link. See also Static test No. 16. [7] Broke No. 4 link. [8] Chain fractured in center of link. See also Static test No. 17.

chain showed no signs of failure after this severe treatment, and the elongation over the links measured 0.65 inch. This chain was then transferred from the drop to the static machine and subjected to the proof load, breaking load, and breaking load plus 40 per cent, after which the load was increased to 452,700 pounds, when the No. 2 link broke, showing solid sections at the fracture.

In dynamic test No. 7, on 2-inch Naco steel chain, the chain was subjected to three blows at 5 feet, three at 10 feet, three at 15 feet, three at 20 feet, 10 at 25 feet, 10 at 30 feet, and 10 at 35 feet. The elongation over

three links was 0.72 inch with no signs of fracture, after which the chain was submitted to the same static loads as test No. 4, and broke through No. 3 link at 467,400 pounds, showing solid metal at the fracture but with a small shrink at one side.

In dynamic test No. 6, on 2-inch navy chain, the chain was subjected to three blows at 5 feet, three at 10 feet, three at 15 feet, three at 20 feet, 10 at 25 feet, and two at 30 feet.

The test was discontinued at this point because of excessive elongation in the chain, which permitted the equalizer to interfere with its support and made it impossible to deliver the full blow to the chain. The elongation over three links was 3.85 inches, and each of the five links was slightly fractured in one or more places. After the third blow at 20 feet the elongation over three links was 1.32 inches, and after the third blow at 25 feet it was 2.08 inches. Undoubtedly this chain was out of commission on account of distortion so far as serviceability is concerned after the third blow at 25 feet.

Results of Static Tests.—Static tests Nos. 13 and 14, on 2¼-inch Naco chain, were made on test specimens having serious shrinkages on one side of the No. 3 link opposite the stud, for the purpose of determining the effect of these defects. In test No. 13 the defective link broke through the defect at 502,400 pounds, and in test No. 14 the No. 5 link broke at 491,900 pounds, showing solid sections at the fracture. The second chain was retested, finally breaking through the defect in No. 3 link at 502,200 pounds. While neither of these chains would have been considered suitable for commercial purposes and the defects would have been readily detected on a casual inspection, it is quite interesting to note the slight effect of the imperfections.

Great Progress Made.—Some reference should be made to our experience in the railway car-coupler field during the past twenty years, for the reason that experience has made possible the accomplishment of results in the cast-steel chain problem in a few weeks which otherwise would have taken several years to attain.

The knuckle of the Master Car Builders' vertical-plane type of coupler has involved a very serious and difficult problem, particularly in the past ten years. Since the adoption of the standard Master Car Builders' contour for that coupler in the early 'nineties, a standard which, because of the requirements of interchangeability, has remained substantially without change up to the present time, there have been enormous increases in locomotive tractive effort, size and weight of car equipment and tonnage hauled per train. The limitations of the Master Car Builders' contour, formed partly by the coupler head and partly by the knuckle, have imposed certain fixed dimensions upon the size of the hub of the knuckle, its weakest point, which have effectually barred any substantial increase in section. The stress to which this hub section is subjected is really enormous, for the knuckle, which is in the form of a hook pivoted at its middle, transmits not only the entire tractive effort of the locomotive in pulling the train, but also receives shocks of great suddenness and intensity, both jerks and buffs, due to the surging of trains, the running out and in of "slack" during application and release of brakes, and in coupling by impact. Dynamometer car records have shown time and again that these pulls and buffs are frequently many times in excess of the tractive effort of the largest modern locomotives.

Notwithstanding the evil effects of knuckle distortion upon proper maintenance of contour, it has been felt for years that nothing but a material with high ductility would be capable of receiving these severe shock stresses. Part of the basis for this feeling has been the theory that a very ductile material would, by its elongation, absorb some of the shock

and so save itself from rupture, but the difficulty with this is that the major part of this elongation is available only after the elastic limit of the material has been passed.

Have Standard Steel Coupler.—Two possible solutions for this problem have presented themselves, the use of a special material with much higher capacity to resist both static and dynamic stresses, and the adoption of an entirely new and heavier coupler with modified contour, having sufficient sectional areas to make resort to special materials unnecessary. The latter method was followed by joint action of all the coupler manufacturers, under the initiative and jurisdiction of the Master Car Builders' Association, and today, after six or seven years' development, we have one standard coupler, somewhat heavier and very much stronger and with its knuckle able to meet the demands of service, though made of ordinary open-hearth steel.

Investigate Material.—In the meantime, realizing the importance of the problem, and the probable length of time involved in developing a standard coupler, we began an exhaustive search for a suitable special material to use in our knuckles. In the early days we started making couplers by the acid open-hearth process, later changing to basic, but our experience with both these steels led us to believe that there was no possibility of increasing the tensile strength sufficiently without loss of ability to withstand shock which would only result in failure to withstand the required Motor Car Builders' drop tests, which tended to compel the use of a soft material. We then began a series of experiments in the electric furnace with high-carbon steels heat treated, and later with alloy steels. Marked improvements in tensile strength were the result, but the ability to withstand shock did not increase and in many cases was diminished. Finally we arrived at a special material to which we later gave the name "Naco" steel, which combined with a high tensile strength extraordinary ability to withstand shock.

This material now used in our coupler knuckles, has raised their elastic limit more than 225 per cent, and their ability to resist shock stresses nearly 100 per cent, over ordinary open-hearth steel. The amount of shock which these knuckles will take without breakage or serious distortion is beyond anything in our experience, and we have been able to develop a specification of our own, along the lines of Motor Car Builders' coupler specifications, but calling for only half the amount of knuckle distortion with the same amount of shock in the drop test, a feature of the greatest advantage in the maintenance of proper contour in service.

Our inspection department is using this Motor Car Builders' drop-testing machine daily in the acceptance or rejection of our knuckle steel heats, and make on the average of 3,000 tests per year. The performance in this machine is borne out in service by the record of more than 1,000,000 knuckles in use without a report from any railroad or from our road service department, of a single failure of a knuckle since the experimental period was passed.

There are so many points of resemblance between the coupler-knuckle problem and that of cast-steel chain as we understand it, that we were able to start at a point which otherwise must have taken years of study and development, with the results set forth. In our opinion, a proper specification for cast-steel chain would require the making of links in dry sand molds or cores, but other details of manufacture should be left optional with the manufacturer. For the chemical requirements, the steel should be made by the electric-furnace process, with phosphorous and sulphur limits of not over 0.04 per cent. The physical requirements should be sufficiently rigid to insure sound material and should embody a dynamic test sufficiently severe to put the ability of the steel to withstand shock

beyond any question of doubt. For this latter purpose we believe the standard Motor Car Builders' drop-testing machine presents the most readily available commercial machine, and a shock test on short pieces of test chain similar to the tests carried out at our Sharon plant seems to be as closely representative of the service to which anchor chain is subjected as any laboratory test can be.—"Engineering."

THE MANUFACTURE AND TESTING OF CAST STEEL CHAIN CABLES.

The following memorandum regarding the manufacture and testing of cast-steel chain cables has just been issued by Lloyd's Register of Shipping:

I.—SUMMARY OF PREVIOUS ATTEMPTS AND PRESENT POSITION.

Owing to the great demand for chain cables and the shortage of wrought iron suitable for their manufacture, suggestions have from time to time been made to manufacture them from steel. The first proposals were to make them from rolled mild-steel bars in a similar manner to that by which wrought-iron cables are made, viz. by welding each link separately by hand. Attempts in this direction had been made many years ago in France, and one of the assistants to this Society's Chief Ship Surveyor witnessed some tests of chains made in this way. The general opinion as to the efficiency of welds in steel is that the higher the tensile strength of steel used, the less reliable the weld will be; and as the security of a chain depends upon the reliability of each and every link in it, the steel chains in question were manufactured from very soft steel. On testing the chains so made, it was found that although the tensile strength of the steel bar used was somewhat in excess of that of good wrought iron, yet the soft steel yielded so much more than iron under stress, that the links of the chain were distorted and locked themselves together, and so rendered the chain useless, long before the breaking stress was reached. The manufacture of steel chain was therefore abandoned at that time.

In December, 1915, two important American firms contemplated making chains of mild steel, and asked the Committee of Lloyd's Register to sanction their use for ship purposes. They stated that steel chains were largely made in the United States, and used for slings and for many purposes other than for ship's cables. After some correspondence, however, the companies interested did not proceed further with their proposals. In October, 1917, the matter was again raised by some American manufacturers, and the Society's surveyors at New York were informed that before steel cables could be accepted for use in classed vessels, crucial tests to prove their suitability would have to be made. The necessary tests were suggested by the Committee, and approval was given for the tests to be made in America and witnessed by the Society's surveyors there. Nothing further appears to have been done as regards making chain cables from rolled-steel bars.

In December, 1917, a representative of a large American company, who was then in this country, stated that his company was proceeding with the manufacture of ship chain cables from cast steel, made by the electric process. He mentioned that by the casting process chain cables could be made in a quarter of the time taken to make ordinary chain with welded links, and that, in addition, there was a great economy of labor by dispensing with that employed in the puddling process of making wrought iron and in rolling the iron into bars. He was given full information as to the approved shape of links and as to other requirements of chain

cables. From a report which has lately been received from the Society's surveyors at New York, it appears that this company submitted to the United States Navy Board six lengths of 2-inch cast-steel chain for testing, three of them being in an annealed condition, and three having been subjected to a special heat treatment. The results of all six tests were nearly identical, the mean showing a breaking strength of 146 tons, the statutory breaking test for iron cables of the same size being 100.8 tons. In these samples the studs appear to have been cast with the links. The excess of strength of the cast steel over the statutory breaking test of wrought-iron chain was 45 per cent, and allowing that good wrought iron chain has a strength of 15 per cent above the statutory test, the advantage of the steel appeared to be 27 per cent above wrought iron. It is noted on the report that the elongation under the proof load was practically nil, showing that the material has a much higher yield point than is customary for chain iron. The steel from which these samples were made had a tensile strength of 33 tons per square inch. Its analysis showed—Carbon .33 per cent, manganese .61 per cent, phosphorus .03 per cent, sulphur .025 per cent, and silicon .145 per cent.

In the circumstances it was considered by the Committee of Lloyd's Register to be advisable to have some experimental cast-steel chains made in this country. A method of molding them by comparatively unskilled labor was devised by the Society's Chief Engineer Surveyor, and one of the British steel-making firms was asked by the Committee to make some samples for testing, steel of different degrees of hardness being employed in the various samples in order to determine the grade of material most suitable for the purpose. Some samples of three different sizes and of different grades of steel were made by the firm, the links having the studs cast in them. The samples were tested at the Netherton Proving House. The results showed that a moderately hard steel, say with a tensile strength of about 34 tons per square inch, is the most suitable material, and that soft steel is not suitable, confirming the conclusions arrived at by the experiments with chain made from soft-steel rolled bar. Some of the links in these specimens were unsound, but the experiments showed that chain cables can be made by casting the links into one another, each link having its stud forming an integral part of itself.

In March, 1918, the Society's surveyors at New York reported by cable that another important American company had developed the manufacture of electric cast-steel cables, and had submitted samples of four different sizes to very severe tensile and shock tests, the results showing the tensile strength to be from 73 to 93 per cent over that required by the statutory test, and the resistance to shock being still greater than these proportions. The results being so satisfactory, the surveyors recommended that the committee's sanction should be given for the use of such cables pending the receipt of a full report. This recommendation was approved by the Committee. A detailed report has now been received, from which it appears that 17 samples in all were tested. Omitting the results of two of the samples, which appear to have been made from soft steel, the tests showed that the breaking strain ranged from 59 to 105 per cent above the statutory test for wrought-iron cables, or from 38 to 78 per cent above the actual strength of high quality wrought-iron cables. In some impact tests made to bend the links crossways, it was found that the deformation was generally less than half that which was sustained by wrought-iron links under the same conditions. In all the tests made, with the exception of one series of cross-breaking tests, the steel chains showed considerable advantage over iron chains of the same size. In the cross-breaking tests in question the results were not strictly comparable, as the iron chains tested had shorter links than the steel ones, and this gave them an

advantage. The full report thus justifies the tentative approval previously given by the Committee for the use of cast-steel cables; and the further question of what will be suitable tests for such cables is dealt with in the second section of this memorandum. It is stated that the Shipping Board of the United States has awarded a contract of the value of one million dollars to the last-mentioned company for cast steel cables.

II.—THE TESTING OF CAST-STEEL CABLES.

The testing of iron chains intended to be used as cables in British ships is regulated by the Anchors and Chain Cables Act, which prescribes the methods of testing, and gives schedules of the strains required to be applied to each size of chain ranging by 32nds of an inch, from 7/16-inch up to 4½-inch diameter in the case of stud link cables, and from ¼-inch up to 1⅝-inch in the case of unstudded link chain. The tests prescribed in the Act have been adopted by the Committee of Lloyd's Register as being suitable for the chain cables for all classed vessels of whatever nationality. The main provisions of the Act are—(1) that a breaking test has to be applied to a sample cut out of each 15-fathom length, the selection of the sample being made by the superintendent, and (2) that the complete length of cable has finally to be subjected to a tensile strain. The Act contemplates that this procedure will be carried out in its entirety after delivery of the cable at a Licensed Proving House.

The breaking strains are devised as tests of the quality of the iron used, and the tensile strains as tests of the workmanship, after the breaking tests have shown the quality of the material to be satisfactory. Whilst the breaking tests deform the links of the test pieces, the tensile tests are arranged to subject the cables to the highest strains which they should withstand without deformation or damage. Less severe strains than those provided for by the schedule would admit of the use of indifferent iron or of careless workmanship, whilst strains on the completed lengths exceeding the statutory tensile tests would be likely to permanently damage the chains, although they might not break them. It is generally recognized that iron chains of ordinary quality have an average strength of about 7½ per cent and high-quality iron chains an ultimate strength of about 15 per cent in excess of the statutory breaking strains. In considering the strength of steel chains it is therefore necessary to compare them with this excess and not with the scheduled breaking tests. As steel chains are so considerably stronger than iron chains the tests suitable for the latter will not be sufficiently severe to discriminate between good and bad steel chains, and it will be necessary to arrange for other tests to be applied.

With regard to the tests of wrought-iron stud-link cables prescribed by the Anchors and Chain Cables Act, it is to be noted that the schedule giving the tensile and breaking strains for the various sizes of cables recognizes that large chains are less reliable in proportion to their size than small chains. As an illustration, the statutory tensile tests applied to all chains up to and including 2½-inch diameter is based upon a stress of 11.45 tons per square inch upon the cross section of the links. For sizes above 2½-inch the tensile test stress is regularly lowered until at the largest size provided for, viz: 4½-inch, it is only 6.9 tons per square inch. As a further illustration, the statutory breaking test for all stud-link cables below 1½-inch is 1.5 times the tensile test, whilst for all such cables above 1½-inch it is only 1.4 times the tensile. In wrought iron the strength is largely dependent upon the amount of work put upon it in making it into bars, and the material of large bars is therefore not so strong as that of smaller bars of the same grade of iron. This is recog-

nized in the British Standard Specifications for various grades of iron which provide for bars over 2 inches diameter or square having one ton per square inch less tensile strength than bars below that size. This difference, however, is very small as compared with the figures in the schedules of test, as a proportionate reduction would only bring the 11.45 tons down to 10.9 tons instead of 6.9 tons per square inch. The great reduction in the relative tensile strain made in the schedule is no doubt due to the fact that, in chain making hand welding has to be employed, and that in the larger sizes the welding cannot be so reliable as in smaller sizes.

Where steel castings are dealt with there is no welding involved, and large links ought to have the same tensile strength per square inch as small ones. In point of fact, their reliability ought to be rather greater, as the effect of an accidental blow hole would be less in a large section than in a small one. Suitable tests for cast-steel cables might be based upon the schedule referred to for cables up to 2½-inch diameter, but for larger cables they should be based upon the sectional areas, and should not be reduced in the same proportion as the schedule reduces tests for iron cables.

As regards the weights of cables, it may be observed that in the case of iron chains, cast-iron studs are fitted, and these are usually made much larger in size than considerations of strength alone would require, and they add to the weight of the cable. In the case of cast-steel chains with the studs forming integral portions of the links much smaller studs will provide for all strength requirements, and the size of stud which will be adopted will depend upon manufacturing conditions, and may be smaller than those of iron cables. It is considered, however, to be advisable, at least for the present, to maintain the minimum weights now required by this Society's Rules.

In all the circumstances, the Committee has, after very careful consideration, come to the conclusion that, bearing in mind the requirements of the Anchors and Chain Cables Act, the following procedure will, as a tentative arrangement, provide a satisfactory method of testing cast steel cables, whatever be the country of manufacture, viz:

(1) The breaking and tensile tests for cables of all sizes should be based upon 22.4 tons and 16 tons per square inch on the section of the links, these stresses being 40 per cent in excess of those required by the Act for iron cables under 2½ inches and over 1½ inches.

(2) It is essential that cast-steel chain cables shall be properly annealed after casting, and this annealing should be the final operation of manufacture. In order to comply with the Act, three links must be selected, and cut out for subjection to the statutory breaking test, and the pieces into which the cable will thus be divided will have to be joined together by means of a new link. This can only be efficiently done on the manufacturers' premises, and as the new or joining link will have to be properly annealed, as well as the rest of the cable, the lengths of the cable must be submitted for inspection and for the selection of the three-link breaking samples by the Superintendent while they are on the manufacturer's premises.

(3) The Superintendent must select a three-link sample from each length, mark this sample, and also the two pieces into which the rest of the cable will then be divided in such a way as will enable him to afterwards identify them. The makers will cast a new link to join the severed portions, and the completed cable and the three-link sample will then have to be annealed simultaneously. The sample will be sent to the Licensed Proving House to be submitted to the breaking test. If this is

satisfactory the completed cable will then be sent to the Proving House to be submitted to the tensile test. Assuming the cable passes this test, the procedure will be at an end and the length of cable will be accepted.

(4.) If either the breaking or the tensile test is unsatisfactory, the following procedure, which is similar to that followed in the case of iron cables, must be adopted:

(*a*) If the breaking test is unsatisfactory, a second three-link sample must be selected by the Superintendent, cut out, and marked. The manufacturer will unite the severed portions by means of one or more new links, and then submit the whole length so united, and also the marked three-link sample, simultaneously, to a proper annealing; after which the selected three-link sample will be sent to the Proving House for further test. If the second three-link sample fails to withstand the prescribed test, the whole length will be rejected; but if it satisfactorily withstands the test, the length of cable will be sent to the Proving House to be subjected to the prescribed tensile strain.

(*b*) If in a length of cable submitted to the prescribed tensile strain one or more links prove to be defective, but not in such a manner as to throw doubt upon the efficiency of the other links of the cable, the defective links may be cut out and the remaining portions of the cable sent back to the manufacturers, who may reunite them by means of new links, after which the reunited cable will have to be again annealed as a whole, and resubmitted to the prescribed tensile test.

(*c*) If more than four per cent of the links fail under the tensile strain, the length must be rejected.

(5.) All the marking of steel cables should be stamped on the studs, not on the main parts of the links. The various marks could be put upon consecutive studs.

These requirements would practically necessitate that the steel foundry shall be within reasonable distance from one of the Licensed Proving Houses.

As is generally known, all cables for British vessels are required to be tested in accordance with the provisions of the Anchors and Chain Cables Act, and it is obviously desirable that the arrangements for testing cast-steel cables should be such as would be sanctioned, whether they are made in the United Kingdom, the United States, or other countries. The Act provides that any test approved by the Board of Trade as a test equal or superior to either the breaking or the tensile test specified in the schedule may be substituted for that test by order of the Board, and the Committee, which has been in communication with the Board of Trade upon the subject, has now received an intimation from the Board that the testing procedure set forth above would be sanctioned for application in this country if the occasion should arise in the near future.—"The Engineer."

MULTIWHIRL OIL COOLER.

Many inquiries about the multiwhirl cooler have reached this desk. The following description from "Power" is quite timely.—Ed.

Among the various devices that have been produced for this purpose [viz: cooling lubricating oil] is the multiwhirl oil cooler, manufactured by the Griscom-Russell Co., 90 West St., New York City.

As shown in Fig. 1, the shell is cylindrical and is made of cast iron. The water head is of cast iron or composition, the floating head is of composition, and the tube plates are of steel. The copper, seamless-drawn tubes are expanded into a fixed tube plate at one end and into a floating head at the other without the use of glands or packing. The

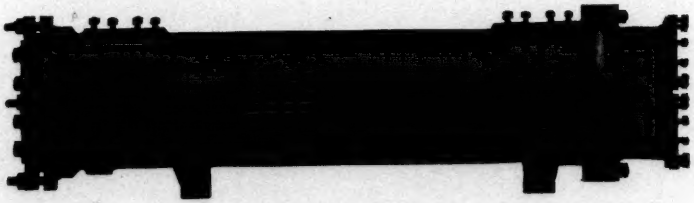


FIG. 1.—MULTIWHIRL OIL COOLER.

FIG. 2.—HELICAL BAFFLE.

floating-head construction permits of the expansion and contraction of the tubes without strain on the joints. As shown in Fig. 2, as well as in Fig. 1, the tubes are arranged in what might be termed a bundle with the baffle surrounding a center piece, so as to make a spiral or helical baffle. This baffle is made of sheet iron or brass perforated to permit the insertion of the copper tubes which constitute the cooling surface. Spacer rods are located near the outer circumference of the baffles in order to secure them rigidly and also to maintain an equal cross-section of flow of the oil through the length of the spiral.

The lubricating oil is circulated through the cooler by means of a pump, flows through the shell and is brought into contact with the tube surface by means of the helical baffle. By reason of this construction the oil-pressure drop is a minimum as the helical baffle guides the oil in its flow without retarding its progress to any appreciable extent. It will be seen that the oil will strike the cooler tube substantially at right angles, and the relatively high velocity of the whirling body of oil scours not only the tube surfaces, but also the shell, thereby tending to keep the interior of the cooler clean. Furthermore, there are no dead pockets in which dirt can accumulate. The cooling water passes through the tubes, which are usually arranged in two or more passes.

In case it is necessary to remove the tube bundle and baffle, it can be done without breaking any oil connection by removing one head and pulling the bundle out of the shell. This can be easily done, as all pipe connections are made directly to the shell itself. By removing the cover plate on the floating head, shown at the left of Fig. 1, access may be gained to the tube's interior without disturbing any other part of the cooler or its connections.

THE 300-H.P. MAYBACH AERO ENGINE.

The appended interesting particulars of the 300-H.P. Maybach engines fitted by the Germans to Rumpler biplanes of the C. 4 type, have been placed at our disposal for publication.

The general distinctive features of the design of the Maybach aero engines are, comparatively speaking, as well known amongst aeronautical engineers in this country as any of the enemy aero engines in service; firstly, by reason of the adoption of the old Wolseley-Maybach aero engines, the design of which was acquired before the war for airship work, and which was based on the design of the original Maybach engine; and, secondly, through the publication of the somewhat incomplete details of construction of the Zeppelin-Maybach engines and their installation, which were collected under great difficulties from the remains of the several Zeppelin airships brought down during the war. These engines, in almost every case, were so badly damaged by fire that it was found almost impossible at the time to compile any really detailed report on their design, or of their power and general performance.

A new and more powerful type of Maybach engine is now being used in service by the enemy, which, according to French reports, develops over 300-H.P., and which possesses great efficiency. During the past few months several of the latest type of Rumpler machines have been captured fitted with the new 300-H.P. Maybach engines. The following report on the design of the 260-H.P. to 300-H.P. Maybach engines is based on a detailed examination of the engine (No. 1,361) taken from a Rumpler two-seater biplane (G. 130), known as the C. 4 type. This machine was brought down in France by a shot which perforated the carburetor and water pump, on January 18, 1918. The machine was, unfortunately, com-

pletely destroyed by fire on landing, but the engine was captured intact and little damaged. After slight repairs this engine has been put into running condition and tested for power, consumption, etc., at R.A.E. The results of these tests are given in the following report, together with metallurgical analyses and mechanical tests of materials and alloys used in the principal parts of the engine.

A complete list of the details of the design is given in the various data at the end of this report, including the percentages of weights of all parts, and the results of separate tests carried out on the carburetor and a new design of petrol pump which is used in conjunction with the interesting high-altitude carburetors.

A brief description of the Rumpler C. 4 and C. 5 type aeroplane, in which these new 300-H.P. Maybach engines are fitted, will, no doubt, be of some interest here. Full details of this machine, however, have already been issued in a published report by the Section Technique de L'Aeronautique Militaire. The Rumpler C. 4 machine is a two-seater biplane, corresponding, more or less, to our D.H. 4, and is generally fitted with a 260-H.P. Mercedes engine. These machines are designed for long-range artillery reconnaissance and photography and are armed with two guns—one Spandau fixed in front of the pilot's seat, firing through the propeller, and a swiveling gun mounted in the observer's seat behind. A two-bladed air screw is fitted, and a semi-circular honeycomb radiator of unusual design is slung above the engine from the center section. These machines are generally flown according to report at high altitudes, *i. e.*, from 15,000 feet to 17,000 feet—until over the lines, and from the French reports, the new 300-H.P. Maybach engines are more flexible and regular in running than the 260-H.P. Mercedes engines, and are generally preferred by the German pilots.

Compared with the 260-H.P. Mercedes engines fitted in these machines, the new Maybach engines are credited with attaining an increase of 200 r.p.m. at altitudes above 2,000 feet, and also possess greater efficiency in speed and climb in the Rumpler biplanes. Total weight of the Rumpler machine fully loaded is approximately 3,439 pounds.

The 300-H.P. Maybach presents several unusual and interesting details, and as compared with the old 240-H.P. Zeppelin-Maybach design, the new engines are undoubtedly a great improvement in general design and efficiency. The quality of the workmanship of every part, including the exterior finish throughout, is exceptionally good, and the working clearances are carried to very fine limits. Compared with any of the types of enemy engines, the workmanship is undoubtedly of a very much more finished nature; every part, nevertheless, shows the usual German characteristics of strength and reliability, combined with standardization of parts and ease of manufacturing, in preference to the saving of weight.

As will be seen by the general views given in Figs. 1 and 2 and by the line drawings, Figs. 14, 15 and 16, on page 869, the general lay-out of the engine follows the usual German six-cylinder vertical type. The compression ratio is exceptionally high, viz: 5.94:1, which necessitates the use of very heavy pistons and connecting rods; in these particulars this engine follows the previous Maybach practice, but the adoption of a cast-iron floating bush in the little end bearing is an interesting development.

Four overhead valves per cylinder are fitted—*i. e.*, two inlet and two exhaust. These are operated by rocker levers mounted on roller bearings in brackets fixed to the cylinder heads, as shown in Fig. 2, and by push rods on each side of the cylinders actuated by separate inlet and exhaust camshafts. The camshafts run in plain bronze bearings in the crank chamber, and are splash lubricated.

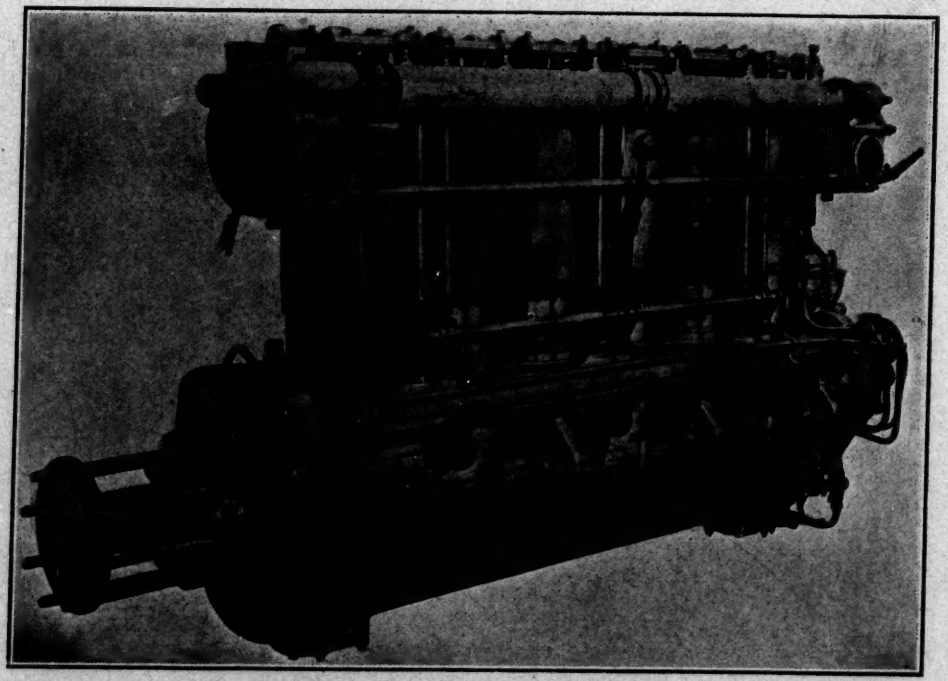


FIG. 1.—INDUCTION SIDE OF MAYBACH ENGINE.

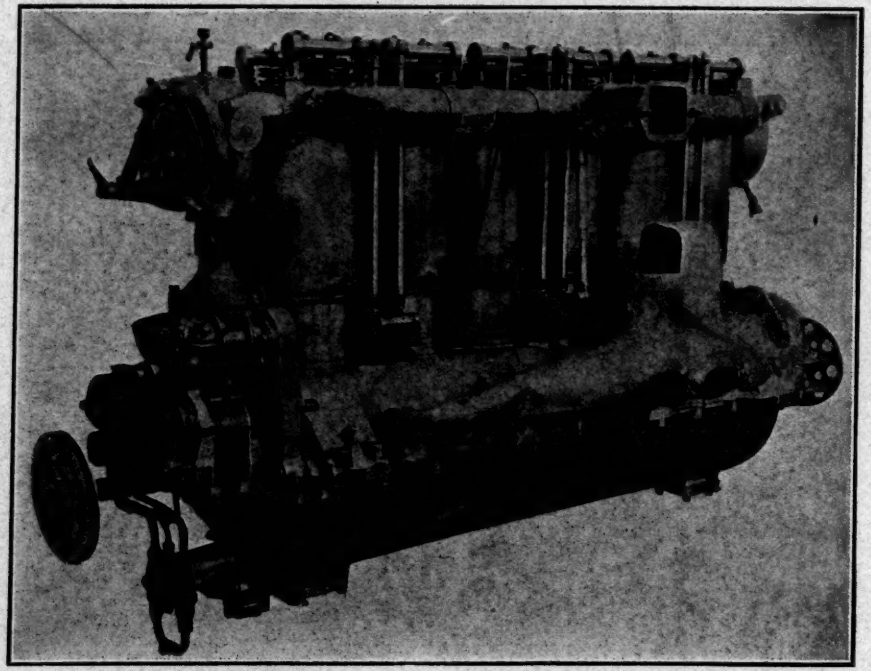
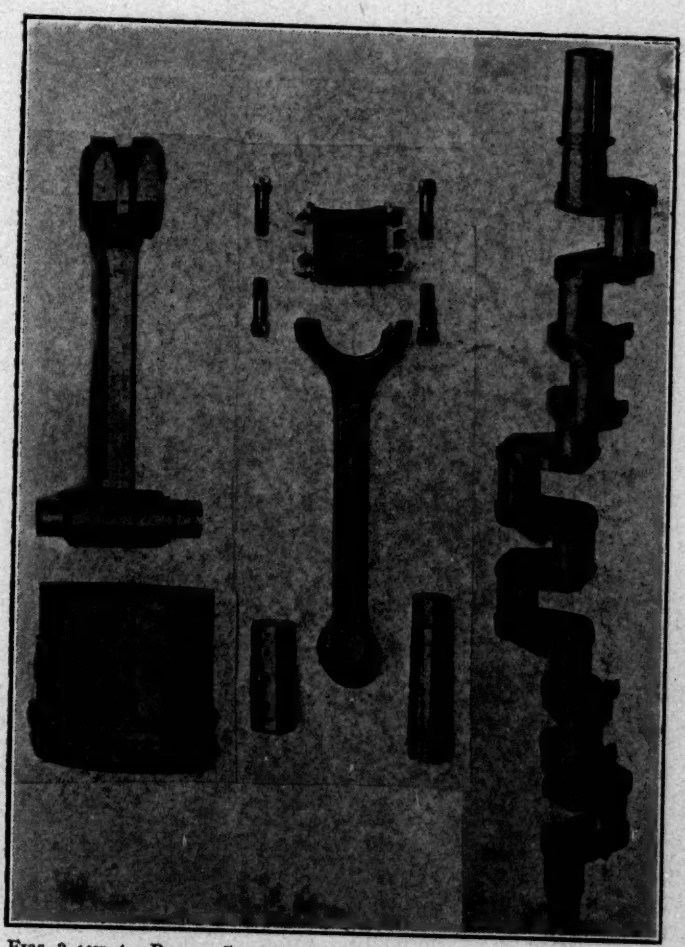


FIG. 2.—EXHAUST SIDE VIEW OF MAYBACH ENGINE.



FIGS. 3 AND 4.—PISTON CONNECTING ROD, ETC. FIG. 5.—CRANKSHAFT.

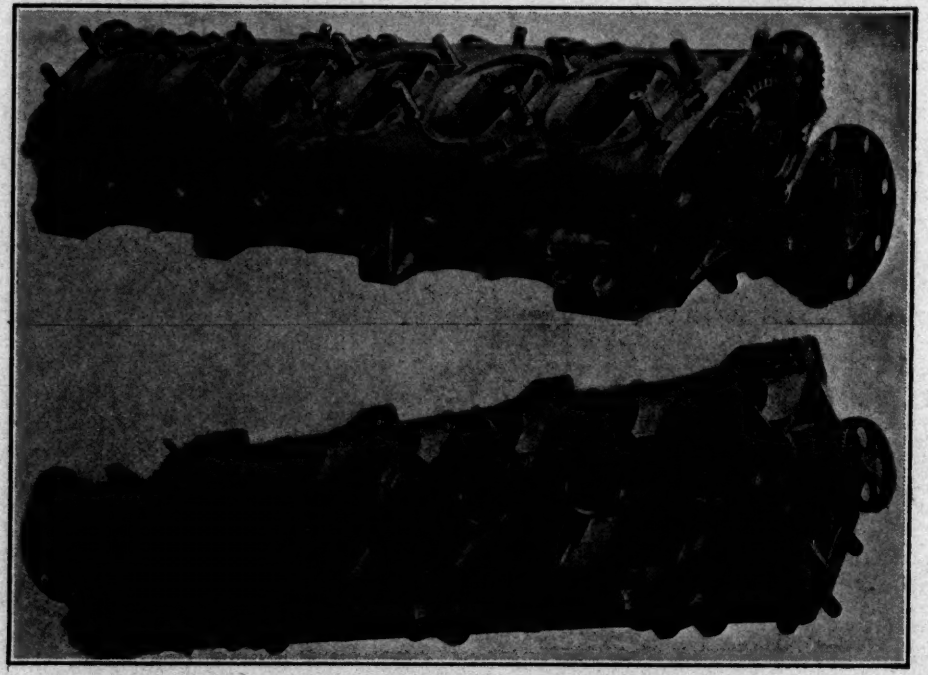


FIG. 6.—TOP HALF OF CRANK CASE.
FIG. 7.—INSIDE VIEW OF CRANK CASE, TOP HALF.

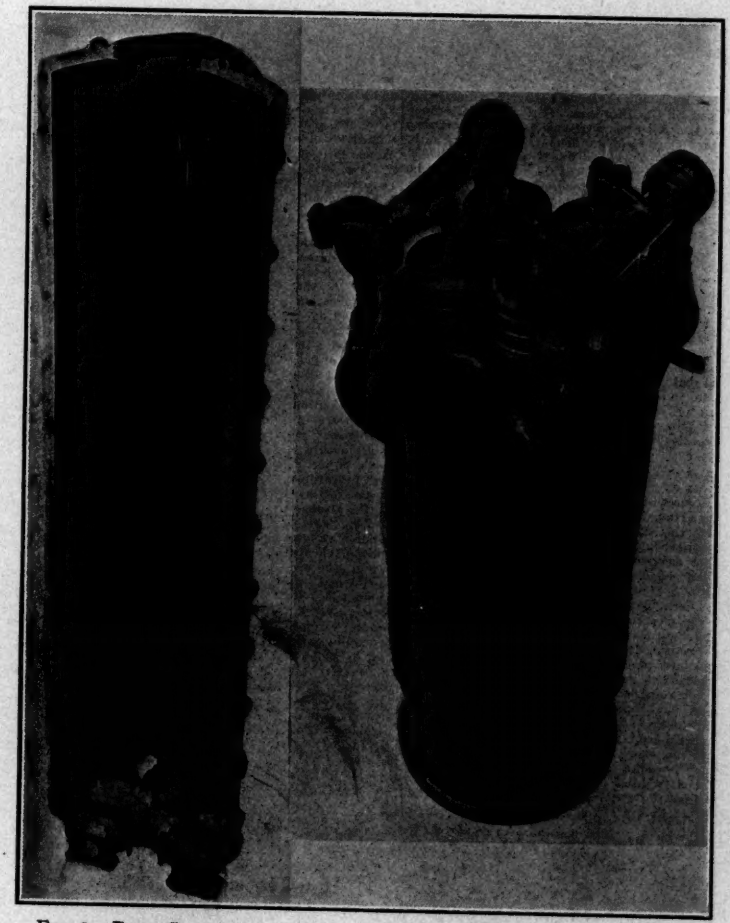


FIG. 8.—BASE CHAMBER, SHOWING OIL PUMPS.

FIG. 9.—CYLINDER COMPLETE.

Little alteration has been made in the design of the massive crankshaft, which runs on plain white-metal lined bearings throughout, and is provided with the well-known Maybach type centrifugal pressure lubrication system to the crank-pins, thence to the gudgeon pins through small pipes inside the hollow connecting rods.

The three separate and detachable-gear oil pumps, which are situated in the bottom of the base chamber, are of new design. The pump-driving shaft at the front end is driven through a ratchet gear on the front scavenger pump, apparently with the object of preventing an air lock in the lubrication system in the event of backfiring.

A double-acting, oil-sealed petrol pump of unusual design is now fitted. This is driven off an extension of the main oil pump spindle at the rear end of the base chamber, and works, of course, in conjunction with the two separate carburetors. These are of the well-known Maybach type, and have been only slightly modified; they are attached, as in the Zeppelin engines, to the front and rear cylinder water jackets. A full description of the functions and workings of this interesting type of carburetor is given towards the end of this report.

The well-known Maybach induction type starting gear is fitted, but is slightly modified in design. The whole of the induction system and the oil-condensing crank-case ventilating system is an interesting point in the design of this engine, as is also the method of attaching the propeller-hub driving flange on the tapered front end of the camshaft.

Practically no alteration has been made to the general design of the very simple and efficient type of water pump, which delivers directly into the bottom of the rear end carburetor water jacket as in the old Maybach engines. Ignition is by two Z.H. 6 type Bosch magnetos, which are driven directly off the rear ends of the camshafts; the position of the sparking plugs in the cylinder heads is a point of interest dealt with in detail in the following description.

Cylinders.—In comparison with the Zeppelin-Maybach engines, the cylinder of the new 300-H.P. Maybach engines are of very much lighter construction; and, as will be seen from the photographs and sectional drawings, are of a wonderfully clean design. Each cylinder shows practically a rectangular view from almost any point of view, with no excrescences or extensions. The whole construction of the cylinders, and particularly the formation of the cylinder head, presents many interesting details of design.

Referring to the sectional drawing, Fig. 10, page 866, it will be seen that each cylinder is built up of a thin steel barrel, machined and ground to 165 mm. bore, and machined on its outer surface to a thickness of 3 mm. The thickness is increased from 3 mm. at the bottom of the water jacket to 4.5 mm. at the cylinder base flange.

Into the top of the cylinder barrel is screwed the cylinder head, which is of cast iron. A detailed sketch of this screwed joint is shown in Fig. 11. The screw thread, it will be noted, is 3.3 mm. pitch, buttress thread. A soft brass washer is fitted at this joint between the cylinder head and the top face of the screwed portion of the cylinder barrel. The bottom of the cylinder barrel extends only 3.5 mm. below the base flange into the registering joint of the crank case, and the bore of the cylinder is very little chamfered.

The design of the cylinder heads and the formation of the water passages round the twin inlet and exhaust valves is unusual, and the double inlet and outlet water connections between each pair of cylinders are of new design. The sparking plug bosses are somewhat inaccessibly situated in the space between the two cylinder-head water joints, as seen in Fig. 9. The two semi-circular bosses which are cast on the sides of the cylinder water joints are provided for the attachment of the two carburetors,

which are fixed to the front and rear cylinders by long studs screwed into these bosses in the water joints. To allow for the interchangeability of the cylinders, these holes are drilled in every cylinder; when the cylinder is not assembled at the front or rear, they are plugged with brass grub screws.

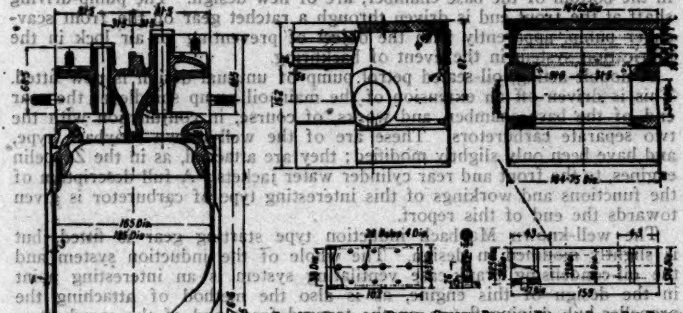


FIG. 12. DETAILS OF PISTON, SPHERICAL PIN AND FLOATING BRASS BUSH.

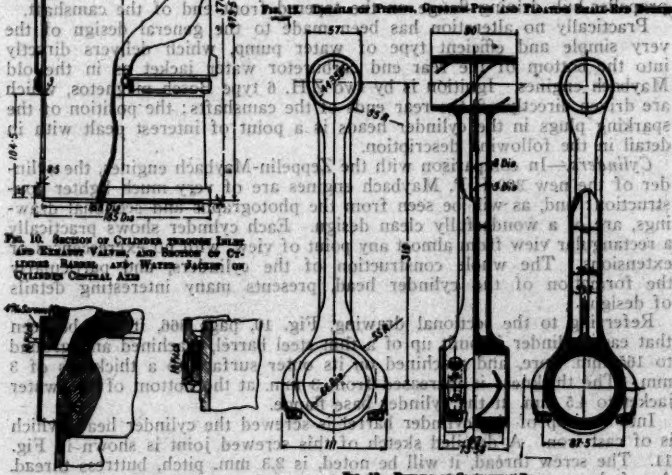


FIG. 10. SECTION OF CYLINDER THROUGH INLET AND EXHAUST VALVES, AND SECTION OF CYLINDER, RADIUS, AND WATER JACKET ON CYLINDER CENTRAL AXIS.

FIG. 13. DETAILS OF CONNECTING ROD.

FIG. 11. DETAILS OF CYLINDER HEAD AND WATER JACKET SCREWED JOINT.

An interesting point of design in these cylinders lies in the fact that no water connections between the cylinders are made other than the twin head joints referred to. (This principle is similar to that used in the cylinders of the Zeppelin-Maybach engines; these were provided with one large water joint, 5 inches diameter, between each pair of water jackets, which was an extension of the cast-iron head. The cylinder barrels were also of cast iron, and were fitted with two inlet and three exhaust valves per cylinder.)

The exhaust-valve pockets are provided with an additional water passage above the exhaust port, and cast so as to give a free water space completely round the center portion of each exhaust valve-stem guide. The valve guides cast in the cylinder heads are fitted with cast-iron bushes pressed into position. The exhaust bushes do not extend the full length of the guides as the inlets do, but are kept back a distance of 10 mm. from the bottom of the guide, so that they are protected from the hot gases. As a further precaution against seizing, the diameter of the bottom of the exhaust bush is slightly reduced so that the bush has room to expand.

Contrary to the usual German practice of building up the water jackets of sheet steel acetylene-welded at the joints the water jackets are machined from cylindrical steel forgings, which, as shown in Fig. 11, are screwed on to the flange machined on the cylinder head; the pitch of this thread is 1 mm. This screwed joint is finally sweated in position with soft solder, the whole depth of the screwed portion having apparently been previously tinned; the joint is locked with four 6 mm. grub screws.

The whole of the water jacket is machined both on the inside and also on the outer surface to a thickness of 1 mm. The water jackets extend to 104 mm. from the base of the cylinders, or, roughly, to two-thirds of the total length of the cylinder barrels; but, as will be noted, the water space is exceptionally narrow—*i. e.*, 7 mm. The only water passages from the cylinder head to the annular water space of the water jackets is through the openings or passages cast in the cylinder heads encircling the sparking-plug bosses, and situated directly below the twin water connections. The water jacket capacity of one cylinder equals 1,384 C.C.S.

Fig. 11 shows an enlarged view of the lower joint between the cylinder barrel and the water jacket. This joint consists of a soft rubber composition packing ring, compressed in position between two steel rings by a large ring which screws on to the bottom of the water jacket by a fine pitch thread in the manner shown in the section. This joint is, to a certain extent, flexible and allows for the relative expansion between the cylinder barrel and the water jacket. Small drain plugs are provided at the bottom of each cylinder water jacket; these are screwed into bosses welded into the water jackets on the exhaust side. The combustion chamber is circular, but it is slightly recessed below all of the four valve seats, which are all of the same diameter—*i. e.*, 54 mm. The area of both the inlet and exhaust ports is 30.15 sq. cm. = 4.66 square inches. The clearance volume of one cylinder = 778.9 cubic cm. = 47.54 cubic inches, giving a compression ratio of 5.94 : 1. Short pieces of aluminum tube are fitted over the top ends of the lugs in the cylinder heads forming the valve-stem guides. These are shown in Fig. 10, and are apparently provided for lubrication purposes. Thin sheet steel plates are bolted to the inlet and exhaust ports of the cylinder heads to stiffen up the engine.

Pistons.—Very little alteration has been made to the general design of the piston in comparison with the smaller Maybach engines. The pistons are of cast iron, and weigh 12.3 pounds each, complete with rings. Four rings are fitted all above the gudgeon pin, the lower one being a scraper ring. All the rings are concentric, and are 6.5 mm. wide (vertically) and 5.5 mm. deep, the depth of the ring grooves being 6 mm. The pistons, as will be seen in Fig. 12, are quite flat on both the top and bottom surfaces of the crown, which is 10.5 mm. in thickness. Eight 4 mm. return oil holes are drilled through the piston below the scraper ring in the usual way. The rings are cut diagonally at 45 degrees, no locating pegs being fitted; and the width of the ring gap in the cylinder is 1.39 mm. = 0.055 inch.

All the piston rings are machine hammered on their inner surfaces. This process which is now extensively used, and is well known, consists

of subjecting the inside face of the ring to a series of mechanical hammer blows at gradually increasing distances apart. The process shows a large number of small transverse depressions extending nearly the width of the ring on the inner surface. These depressions caused by the hammer are approximately 1 mm. apart at the back portion of the ring—*i. e.*, the part of the ring farthest from the gap—and increase in mathematical progression to about 4 mm. apart towards the front of the ring up to about 1 inch, on either side of the gap. This mechanical hammering of the inside ring has the effect of giving the desired uniform pressure to the concentric ring against the cylinder wall.

The holes for the gudgeon pin are bored eccentrically in the gudgeon-pin bosses, giving a thickness of metal of 14 mm. above and 9 mm. below the gudgeon pin. An oil groove is cut in the piston skirt below the gudgeon pin, evidently to assist the scraper ring and the piston lubrication. The inside of the skirt is machined up as far as the bottom of the gudgeon-pin bosses, and a beveled lip is turned on the bottom flange of the skirt. The inside of the piston is also machined flat across the head, and down as far as the top of the gudgeon-pin bosses with a 15 mm. radius; an annular rib is cast on the inside of the skirt to reinforce the gudgeon bosses. Other details of the design of the pistons are given in the sectional drawings, Fig. 12.

Gudgeon Pins.—The gudgeon pins are 159 mm. long, slightly beveled at the end, and are fixed in position in the piston by a single cheese-headed 9 mm. set screw in the usual way and locked with a split pin. The gudgeon pins are hardened and ground to three parallel outer diameters, being made of smaller diameter at one end than the other. The center portion which makes the small end bearing is 38 mm. diameter, and the two ends which fit into the gudgeon-pin bosses are 37.5 mm. and 38.25 mm. diameter respectively. The pins are bored 23 mm. diameter in the center, tapering to 28 mm. at each end.

Floating small end bushes are now fitted to the gudgeon pin in the connecting rod smaller ends. These floating bushes are made of cast iron of a very soft nature. The effective bearing length of the floating bushes is 93 mm., and their external diameter is 44.3 mm. For lubrication purposes a number of 4 mm. holes are drilled and countersunk on the outside. The projected area of the small end bearing surface on the gudgeon pin = 35.35 sq. cm., and the bearing surface in the connecting rod small end = 39.9 cm.

The weight of each gudgeon pin = 1.75 pounds, whilst the cast-iron floating bushes weigh 0.62 pound each. Details of the gudgeon pins and the floating connecting rod small end bushes are given in Fig. 12, on page 866, and in Fig. 4.

Connecting Rods.—Following the usual Maybach design, the connecting rods are of square section beveled at the four corners, and bored up the center from the big end with a 38-mm. diameter hole, which is screwed and plugged as shown in Fig. 13, page 866. The big end bearing cap is secured by four 14-mm. bolts, which are screwed 1.5 mm. pitch into the top half of the connecting-rod big ends. The inner surface of the big-end bearing shells are machined with a screw-cutting tool which leaves a fine pitch screw thread. This provides a serrated bedding for the white-metal lining. No locking nuts are fitted to the big-end bolts, but the square heads of the bolts are drilled, and prevented from becoming unscrewed when in position by a 4-mm. diameter pin, which is long enough to pass through the heads of each pair of bolts. Two helical grooves, 2 mm. wide, are cut in the big-end bearings. These oil grooves intersect each other on the center of the bottom half of the bearing as shown in Fig. 4; the oil ways cut in the top half of the big-end bearing metal are only taken half way on each side.

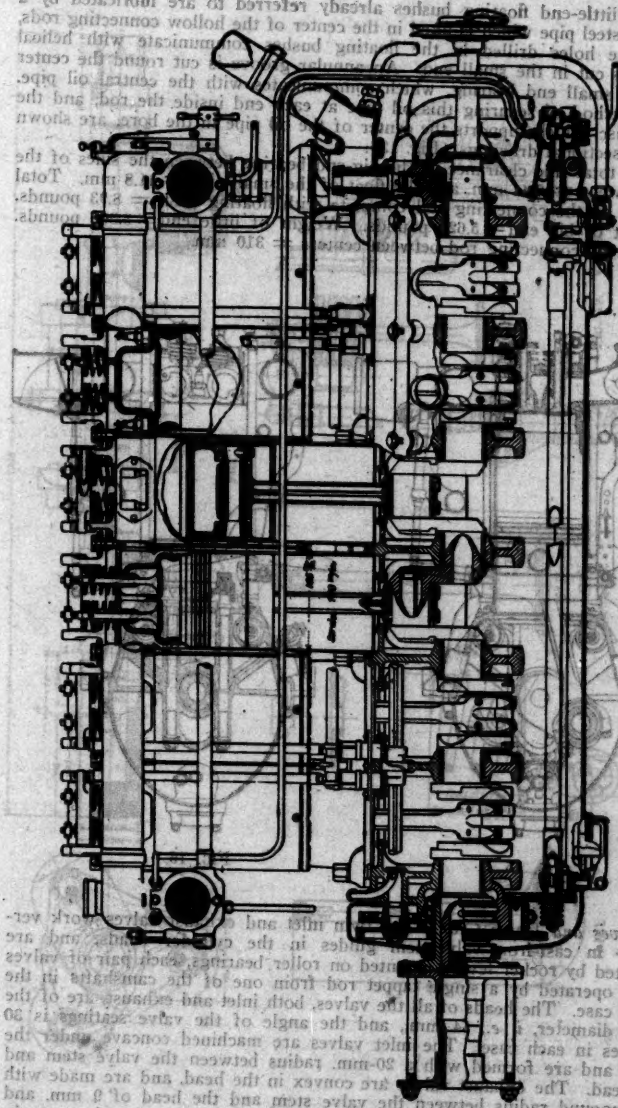


FIG. 14.

The little-end floating bushes already referred to are lubricated by a 5-mm. steel pipe which is fixed in the center of the hollow connecting rods, and the holes drilled in the floating bushes communicate with helical grooves cut in the small end. An annular groove is cut round the center of the small end bearing, which communicates with the central oil pipe. The method of securing this oil pipe at each end inside the rod, and the steel disc, which supports the center of the oil pipe in the bore, are shown in the sectional drawings (Fig. 13).

The total side clearance of the big-end bearing between the sides of the crank-webs = 0.44 mm. and the float of the small end = 11.8 mm. Total weight of the connecting rod complete with floating bush = 8.93 pounds. Weight of big end = 5.625 pounds. Weight of little end = 3.305 pounds. Length of connecting rod between centers = 310 mm.

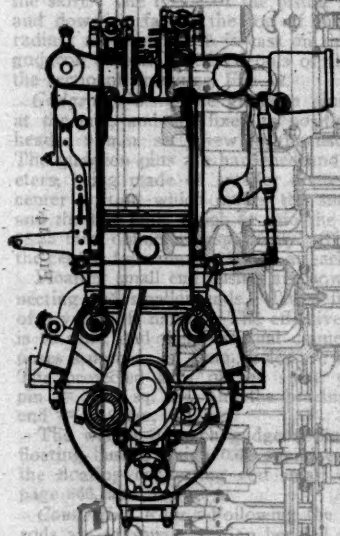


FIG. 15.

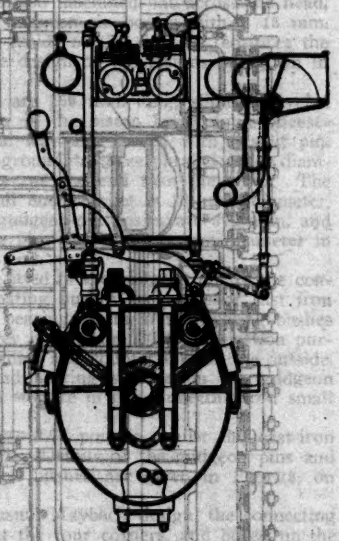


FIG. 16.

Valves and Valve Gear.—The twin inlet and exhaust valves work vertically in cast-iron valve-stem guides in the cylinder heads, and are operated by rocker levers mounted on roller bearings, each pair of valves being operated by a single tappet rod from one of the camshafts in the crank case. The heads of all the valves, both inlet and exhaust, are of the same diameter, *i. e.*, 54 mm., and the angle of the valve seatings is 30 degrees in each case. The inlet valves are machined concave under the head, and are formed with a 20-mm. radius between the valve stem and the head. The exhaust valves are convex in the head, and are made with a compound radius between the valve stem and the head of 9 mm. and 25 mm. The diameter of all the valve stems is 11 mm., the clearance in

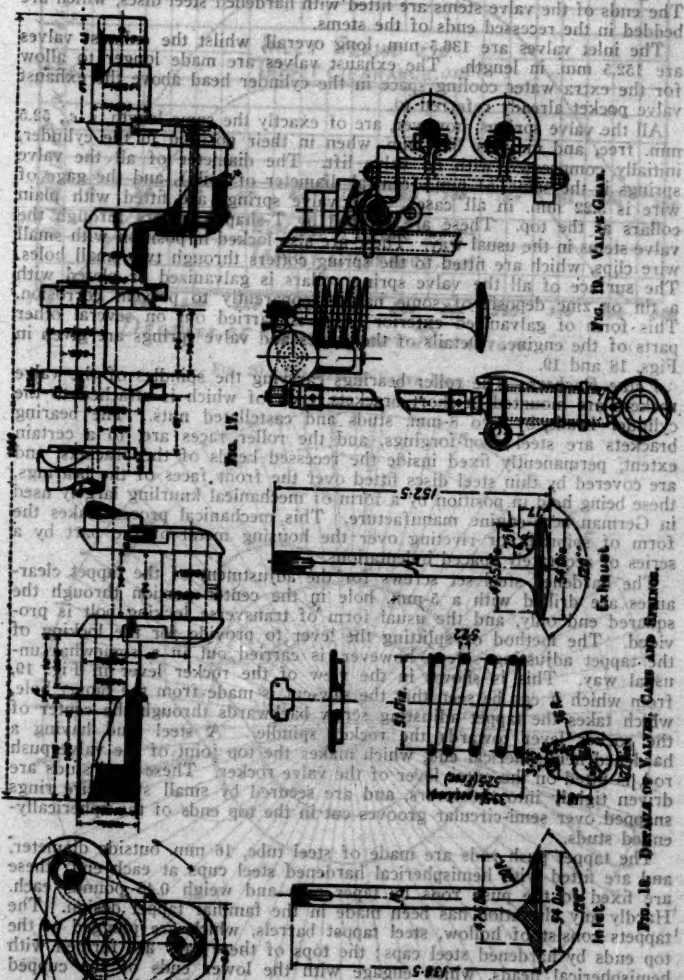


FIG. 15.—Valve Gear.

FIG. 16.—Details of Valves, Cams and Springs.

the guide bushes being 0.12 mm. (inlet) and 0.15 mm. (exhaust) (cold). The ends of the valve stems are fitted with hardened steel discs, which are bedded in the recessed ends of the stems.

The inlet valves are 136.5 mm. long overall, whilst the exhaust valves are 152.5 mm. in length. The exhaust valves are made longer to allow for the extra water cooling space in the cylinder head above the exhaust valve pocket already referred to.

All the valve springs, however, are of exactly the same length, *i. e.*, 52.5 mm. free, and measure 39.5 mm. when in their position in the cylinder, initially compressed with no valve lift. The diameter of all the valve springs is the same, 51 mm. (central diameter of coils), and the gage of wire is 5.22 mm. in all cases. The valve springs are fitted with plain collars at the top. These are secured by T-shaped cotters through the valve stems in the usual way. These are also locked in position with small wire clips, which are fitted to the spring cotters through two small holes. The surface of all the valve spring collars is galvanized or plated with a tin or zinc deposit of some nature, apparently to prevent corrosion. This form of galvanized exterior finish is carried out on several other parts of the engine. Details of the valves and valve springs are given in Figs. 18 and 19.

Valve Rockers.—The roller bearings carrying the spindles of the valve rockers are mounted in short brackets, each of which is attached to the cylinder heads by two 8-mm. studs and castellated nuts. The bearing brackets are steel drop-forgings, and the roller races are, to a certain extent, permanently fixed inside the recessed heads of the brackets, and are covered by thin steel discs fitted over the front faces of the bearings, these being held in position by a form of mechanical knurling largely used in German aero-engine manufacture. This mechanical process takes the form of spinning or riveting over the housing metal of the part by a series of accurately spaced indentations.

The hardened steel set screws for the adjustment of the tappet clearances are drilled with a 5-mm. hole in the center portion through the squared end only, and the usual form of transverse locking bolt is provided. The method of splitting the lever to provide for the locking of the tappet adjusting screw, however, is carried out in a somewhat unusual way. This is shown in the view of the rocker lever in Fig. 19, from which it can be seen that the saw-cut is made from the tappet hole, which takes the tappet adjusting screw backwards through the center of the rocker lever towards the rocker spindle. A steel stud having a hardened hemispherical end, which makes the top joint of the valve push rod, is fitted on the outer lever of the valve rocker. These steel studs are driven tightly into the levers, and are secured by small steel wire rings snapped over semi-circular grooves cut in the top ends of the spherically-ended studs.

The tappet push rods are made of steel tube, 16 mm. outside diameter, and are fitted with hemispherical hardened steel cups at each end; these are fixed to the push rods by taper pins and weigh 0.48 pounds each. Hardly any alteration has been made in the familiar tappet design. The tappets consist of hollow, steel tappet barrels, which are plugged at the top ends by hardened steel caps; the tops of these caps are formed with hemispherical heads, which engage with the lower ends of the cupped push rods. The hardened cam rollers are held in position by the grooves machined in the bottom ends of the tappet barrels; these grooves are made slightly more than a semicircle to prevent the roller from falling out. Side movements of the rollers is prevented by the tappet guides. The cam rollers are 15 mm. in diameter, and are bored with a 4.5 mm. hole. Four small holes are bored radially in the sides of the hollow tappet

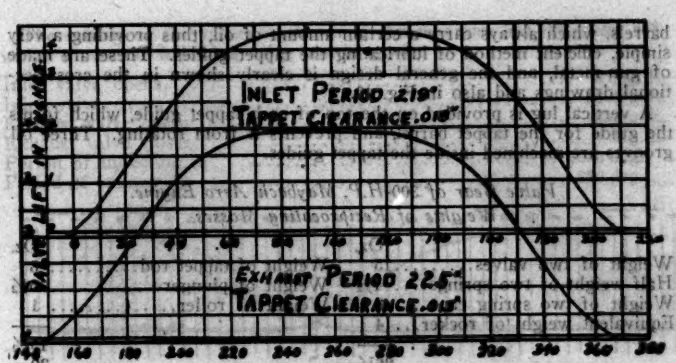


FIG. 20. DIAGRAM OF VALVE LIFT.

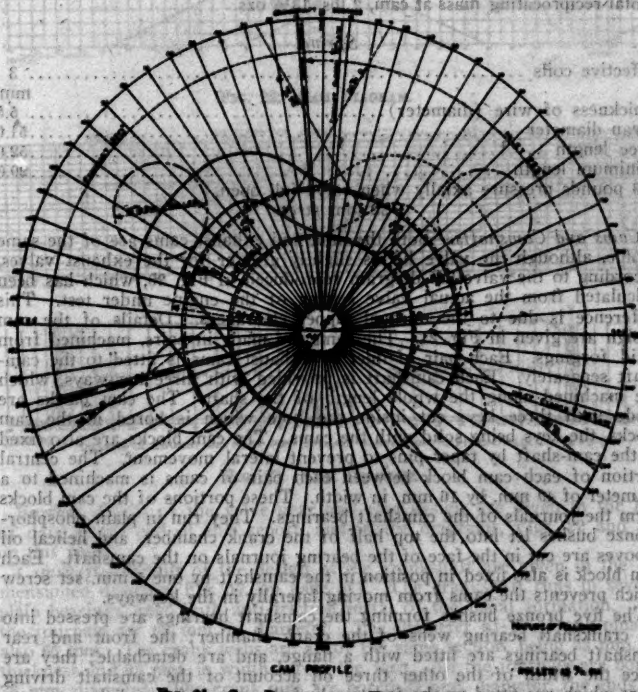


FIG. 21. CAM DIAGRAM AND VALVE TIMING.

barrels, which always carry a certain amount of oil, thus providing a very simple, efficient method of lubricating the tappet guides. These are made of gun-metal, and the general design is clearly shown in the cross-sectional drawings and also in Fig. 19.

A vertical lug is provided on the top of each tappet guide, which forms the guide for the tappet barrel, and prevents it from rotating. Three oil grooves are machined inside the tappet guides.

Valve Gear of 300-H.P. Maybach Aero Engine.

Weights of Reciprocating Masses.

	Oz.		Oz.
Weight of two valves	15	Weight of tappet rod	8
Half weight of two springs	4½	Weight of plunger	6½
Weight of two spring caps	3	Weight of roller	1
Equivalent weight of rocker	4		
	26½		15½
			30
			45½

Multiply by ratio of 35 : 31 = 30 ozs.
Total reciprocating mass at cam, 2 lbs. 13½ ozs.

Spring.

Effective coils	3
	mm.
Thickness of wire (diameter)	5.5
Mean diameter	51.0
Free length	52.0
Minimum length	29.0

133 pounds pressure axially when valve full open.
32 tons stress.

Cams and Camshafts.—Both the inlet and exhaust cams are of the same profile, although the valve lift is slightly greater for the exhaust valves, according to the valve-lift diagram reproduced in Fig. 20, which has been calculated from the actual valve setting of the engine under test. This difference is due to the unequal tappet clearances. Details of the cam design are given in Fig. 21. The cams are made in pairs, machined from solid forgings. Each pair of inlet or exhaust cams is fitted to the cam-shaft separately. Each camshaft is provided with three key-ways, which are machined along the whole length of the shaft. The cam blocks are made with three keys cut inside the hole which is bored in the cam blocks, the keys being solid with the cams. The cam blocks are also fixed to the cam-shaft by taper pins to prevent lateral movement. The central portion of each cam block between each pair of cams is machined to a diameter of 49 mm. by 16 mm. in width. These portions of the cam blocks form the journals of the camshaft bearings. They run in plain phosphor-bronze bushes let into the top half of the crank chamber, and helical oil grooves are cut in the face of the bearing journals on the camshaft. Each cam block is also fixed in position in the camshaft by one 5 mm. set screw which prevents the cams from moving laterally in the keyways.

The five bronze bushes forming the camshaft bearings are pressed into the crankshaft bearing webs of the crank chamber; the front and rear camshaft bearings are fitted with a flange, and are detachable; they are twice the width of the other three on account of the camshaft driving gears which are fitted on the front end of the camshafts, and the magneto driving gears, etc., which are fitted to the rear ends of the camshafts.

The setting of the valves is shown in the diagram, Fig. 21, page 873, which shows the following valve positions, viz:

Inlet valve opens	8 deg. E.
Inlet valve closes	35 deg. L.
Exhaust valve opens	33 deg. E.
Exhaust valve closes	7 deg. L.
Period of induction	223 deg.
Period of exhaust	220 deg.

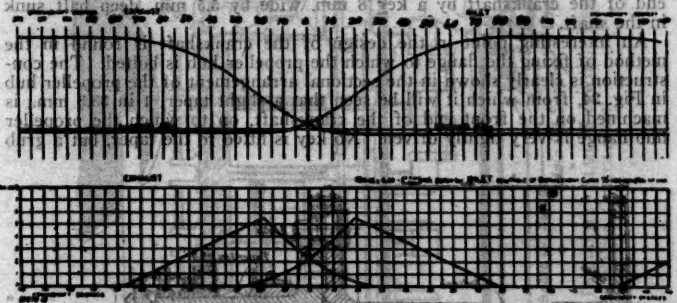


FIG. 22. CAM DIAGRAM.

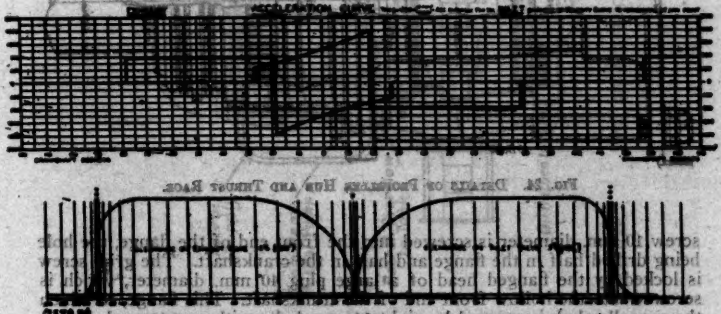


FIG. 23. CAM DIAGRAM.

Further details of the valve gear and cam design are given in the list of data at the end of this report, and in the diagrams, Figs. 22 and 23, on this page.

Camshaft.—Details of the design of the crankshaft are given in the dimensioned drawing, Fig. 17, on page 871. The crankshaft runs in seven plain bearings lined with anti-friction metal of the usual formation. All the journals are 66 mm. in diameter. The diameter of the crank-pins is also 66 mm. The crank webs are all of the same section, being 23 mm. in thickness and 95 mm. in width across the face. All the journals are bored 36 mm. diameter. The crank pins are bored 39 mm. diameter, the ends of the holes being plugged with the centrifugal oil scoops characteristic to the Maybach engines. (Details of these are dealt with later in the notes

oil lubrication). The rear end of the crankshaft carries the extension shaft, on which is mounted the wireless dynamo driving pulley, which embodies a friction clutch. This design of wireless pulley is a standard German engine fitting.

The extension shaft referred to above is a driving fit in the rear end of the crankshaft, which is bored 42 mm. the extension shaft is locked by a 7-mm. taper pin. Two centrifugal oil-thrower rings are machined on the extension shaft. These details are shown in the general arrangement of the engine in Fig. 14. The main driving bevel gear is fixed to the rear end of the crankshaft by a key 8 mm. wide by 5.5 mm. deep half sunk in the shaft.

An interesting point in the design of the crankshaft is found in the method of fixing the flange to which the propeller hub is bolted. The construction is clearly shown in the sectional arrangement of the propeller hub in Fig. 24, from which it will be seen that a slight taper, 1 in 33.3 mm., is machined on the front end of the crankshaft, on to which the propeller hub flange is very tightly driven. No key is fitted to the taper, but a grub

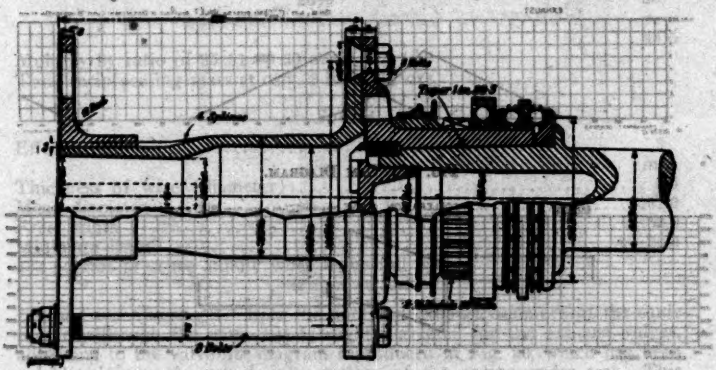


FIG. 24. DETAILS OF PROPELLER HUB AND THRUST RACE.

screw 10 mm. diameter is screwed into the front end of the flange, the hole being drilled half in the flange and half in the crankshaft. The grub screw is locked by the flanged head of a large plug 40 mm. diameter, which is screwed into the hollow front end of the crankshaft. This flange, to which the propeller hub is secured by eight 14-mm. bolts with countersunk heads, is mounted on a large single-race ball-bearing 132 mm. outside diameter, behind which is fitted the double-thrust ball race, 108 mm. diameter on the ball center diameter.

On the front portion of this propeller hub carrying flange is mounted a double oil-thrower ring, which is driven on to the outer diameter of a split collar or bush as shown in Fig. 24. The driving gear for the cam-shafts is machined on this propeller hub flange; the teeth of the gear are integral with the flange, hence the splitting of the bush which carries the double oil-thrower rings.

Propeller Hub.—The general details of the construction of the propeller hub are given in Fig. 24. Eight 15-mm. bolts are used to bolt the front flange to the two rear propeller hub flanges, and the propeller is secured by castellated nuts in the usual manner. The surface of the propeller hub and flanges is galvanized with deposited tin to prevent corrosion. The

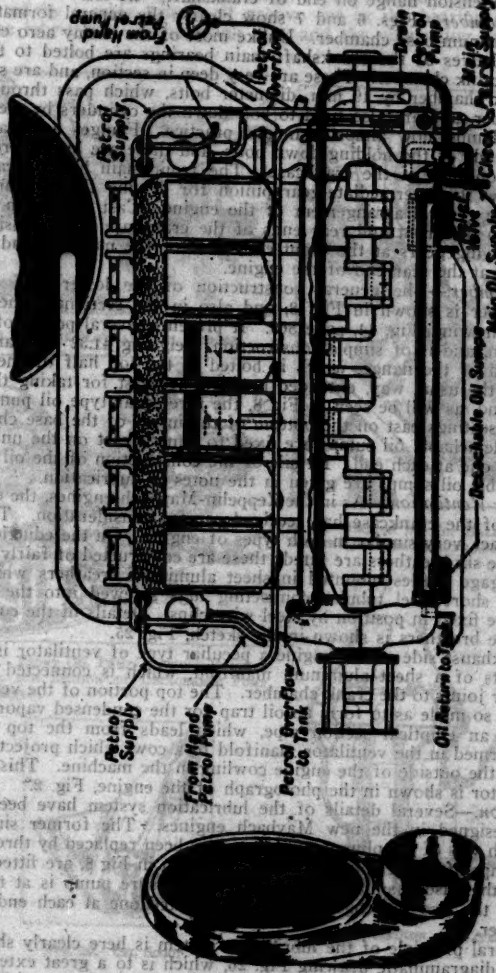


FIG. 26.—LUBRICATION DIAGRAM; ALSO SHOWING PETROL SYSTEM AND WATER CIRCULATION.

FIG. 25.—CRANK-CASE BREATHER.

front flange floats on the propeller hub in four castellations, 17 mm. wide by 3 mm. deep. The total weight of the complete propeller hub, with all bolts, less extension flange on end of crankshaft,=21 lbs.

Crank Chamber.—Figs. 6 and 7 show clearly the general formation of the cast aluminum base chamber. Unlike most of the enemy aero engines, the bottom halves of the crankshaft main bearings are bolted to the top half of the crank chamber. These are very deep in section, and are secured to the crank chamber by 19 mm. diameter bolts, which pass through the crank chamber, and are also used to bolt down the cylinders by means of triangular clamps, now almost standard practice. Flanges are machined on the top ends of the holding-down bolts, and these are let into recesses cut in the top face of the crankcase. The front main bearing cap also carries below the intermediate-gear pinion for driving the oil pumps as shown in the general arrangement of the engine. Cast aluminum covers are bolted to the front and rear ends of the crankcase for enclosing the camshaft driving gears at the front end, and for the magneto and interrupter gears at the rear end of the engine.

Base Chamber.—The general construction of the lower half of the crank chamber is shown in Fig. 8, and also in the sectional general arrangement drawing, Fig. 14, page 869. It presents several points of interesting design, and is of simple construction, weighing 41.32 lbs. bare. At the rear end of the flange, which is bolted to the top half of the crank chamber in the usual way, extended brackets are cast for taking the two magnetos, and, as will be seen in Fig 8, the three-gear type oil pumps are situated on seatings cast on the bottom of the inside of the base chamber. The small detachable oil sumps are fixed to flanges cast on the underside of the base, one at each end. Details of the construction of the oil pumps and detachable oil sumps are given in the notes on lubrication.

Crankcase Ventilation.—As in the Zeppelin-Maybach engines, the efficient ventilation of the crankcase has received careful consideration. The design is, in fact, very similar in both types of engine. On the eduction side of the engine six breathers are fitted; these are constructed of fairly coarse brass wire-gage baffles mounted in sheet aluminum breathers which are attached to short steel tubular connecting lugs screwed into the crankcase, and are fixed in position by steel wire clips. Details of the construction of these breathers is shown in the sketch, Fig. 25.

On the exhaust side of the engine a peculiar type of ventilator is fitted. This consists of a sheet aluminum manifold, which is connected by six short rubber joints to the crank chamber. The top portion of the ventilator manifold is so made as to form an oil trap for the condensed vapor. This consists of an elliptical section pipe, which leads from the top of the chamber formed in the ventilator manifold to a cowl which projects backward from the outside of the engine cowling in the machine. This crankcase ventilator is shown in the photograph of the engine, Fig. 2.

Lubrication.—Several details of the lubrication system have been completely redesigned in the new Maybach engines. The former single oil pump, which was of the plunger type, has now been replaced by three separate oil pumps of the gear type. These, as shown in Fig 8, are fitted in the bottom of the base chamber. The main oil-pressure pump is at the rear end, whilst the two scavenger pumps are situated one at each end of the base chamber.

The general principle of the lubrication system is here clearly shown in the special diagrammatic drawing Fig. 26, which is to a great extent self-explanatory. In this diagram the lubrication system is shown with all the oil pipes and oil-ways marked in black. The system briefly described is as follows: Oil is delivered from the separate oil tank by the pressure oil pump, which is the rearmost pump of the two at the rear of the sump, and forces oil under pressure to the crankshaft journal bearings through an external oil main, which runs the whole length of the crankcase on the induction side, as shown in Fig. 27.

From this main oil pipe oil is forced to the journal bearings through oil ways drilled diagonally in the crank-chamber casting through the transverse webs which support the housings of the journal bearings; these are lubricated by a wide helical oil groove cut in the bearing white metal. For the lubrication of the crank pins and the connecting rod small-end

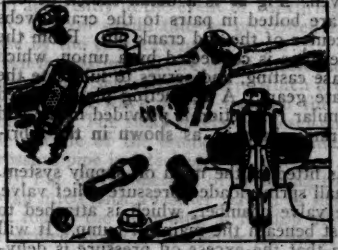


FIG. 27. OIL MAIN AND OIL FILTERS.

FIG. 30. SECTION OF TOP HALF OF CRANK-CASE SHOWING JOURNAL BEARING AND OIL PASSAGE.

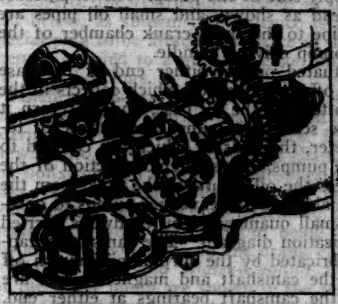


FIG. 28. PISTON OIL PUMP SHOWING RATCHET DRIVE AND SECTION OF DETACHABLE PUMP IN CRANKCASE.

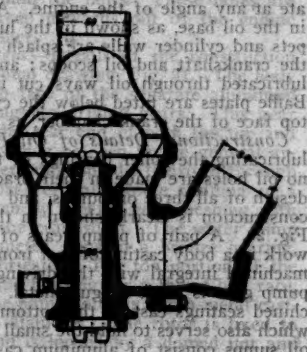


FIG. 31. SECTION OF WATER PUMP.

FIG. 29. OIL SCOOP.

bearings, the well-known Maybach system of centrifugal pressure lubrication is adhered to. In this system the oil, which is forced out through the ends of the journal bearings, is collected by the oil scoops, which are bolted to the outer sides of each crank web, as shown in the view of the crank-shaft already referred to.

By centrifugal action the oil is forced up the outer surface of each crank web and led through a channel into the hollow crank pins, thence through a small hole bored radially in each crank pin to the connecting rod big-end bearings. From the big-end bearings the oil is forced upwards to the small-end bearings, with their floating gudgeon-pin bushes of cast iron, through the steel pipe fixed in the center of the connecting rods. Details of this design have already been given. Fig 29 is a detail sketch of one of the aluminum oil scoops, which are bolted in pairs to the crank webs by a bolt which passes through the center of the end crank pin. From the front end of the main oil pipe a steel pipe is connected by a union, which leads to a hole drilled in the crankcase casting, and serves to lubricate the bearings of the camshaft intermediate gears. A restricting plug is fitted into the end of this oil lead, and a similar restriction is provided in the end of the main leading back into the crank chamber, as shown in the lubrication diagram.

A check valve, it will be noticed, is fitted in the main oil supply system. This check valve, together with a small spring-loaded pressure relief valve, is situated inside a small detachable valve chamber, which is attached to the bottom of the crank chamber just beneath the main oil pump. It will be noticed in the lubrication diagram that the excess oil pressure is delivered back into the oil pipe on the supply side of the pump. An oil pressure gage is connected to the main oil lead as shown, and small oil pipes are also taken from the main pressure pipe to the small crank chamber of the petrol pump, and also to the water-pump driving spindle.

The two scavenger oil pumps, situated one at either end of the base chamber, are designed so as to draw off the return oil which collects in the small detachable oil sumps in the bottom of the oil base, and to return it to the tank from either end. These scavenger pumps are connected together by a steel pipe, 21 mm. diameter, the ends of which are coupled to the suction ports of the two gear oil pumps. This inter-connection of the two pumps to a certain extent balances the oil return to the tank from the front and rear ends of the oil base, enabling both scavenger pumps to operate at any angle of the engine. A small quantity of oil is always carried in the oil base, as shown in the lubrication diagram. The camshafts, tappets and cylinder walls are splash lubricated by the surplus oil thrown off the crankshaft and oil scoops; and the camshaft and magneto gears are lubricated through oil ways cut in the camshaft bearings at either end. Baffle plates are fitted below the cylinders in the recesses machined in the top face of the crank chamber, as shown in Fig. 30.

Constructional Details of Oil Pumps, etc.—No provision is made for lubricating the roller bearings of the overhead valve-rocker spindles, and no oil holes are made in their brackets for hand lubrication. The general design of all three oil pumps, and their delivery is the same. The general construction is clearly shown in the sketch of the front scavenger pump, Fig. 28. A pair of pump gears of the usual formation, 30 mm. diameter, work in a body casting of cast iron. The top pump gear is of steel, and is machined integral with the driving spindle of the pump, whilst the lower pump gear is made of gun metal. The oil pumps are secured to the machined seatings cast on the bottom of the oil base by a stud in each case, which also serves to hold the small detachable oil sumps in position. These oil sumps consist of aluminum castings, into the cupped center of which is screwed a spun-brass oil trap, formed as shown in the sectional perspective sketch. These oil sumps are connected to the suction ports of the oil pumps by a passage formed in the sump body castings.

All three oil pumps are driven at the same speed, the two scavenger pumps being connected together by a light tubular shaft which engages with dog clutches at either end; the pressure oil pump is driven off the spindle of the rear scavenger pump through a dog clutch extension of the spindle. The method of driving the oil pumps at the front end of the engine is of

interesting and unusual design. As shown in the sketch, Fig. 28, the driving-gear wheel, which is mounted on the front extension of the pump spindle, is provided with a spring-loaded face ratchet, the teeth of which are cut inside the gear as shown, and engage with similar teeth cut on a flange on the front of the pump-driving spindle. The strength of the spring is such as to allow the driving gear to overrun the driving spindle, and apparently in the event of backfiring, all the oil pumps become inoperative, and oil would thus be prevented from being sucked back into the oil base by the two scavenger pumps.

An interesting feature in the lubrication system of these engines is found in the small separate oil filters which are provided at the top of each of the diagonal oil ways drilled through the webs of the main journal-bearing housings. The small filters are screwed into the bosses, which are cast on the outside of the crank chamber, and to which the detachable main oil pressure pipe is attached.

In Fig. 27 the method of attaching the main oil pipe by a set screw and a cap fitted with a leather washer is shown. The set screw is screwed into the top of the filter plug at the top of the drilled oil lead, and a small jet or drilled plug is fitted below the set screw, the plug being let into a small recess cut in the top of the oil lead. The bore of this plug corresponds to the bore of the restriction formed in the oil lead drilled in the filter plug just below, as shown. This device is apparently provided to allow for a special plug to be attached in place of any of the six set screws, which can then be coupled to a pressure gage, thus giving the oil pressure at any of the main bearings.

Further details of the formation of the oil passages drilled through the crank case to the journal bearings and also of the construction of the main bearing cap are given in the sketch, Fig. 30, which shows the underneath view of the top half of the crank chamber. The oil is led round the bearing bolts through circular oil grooves cut in the face of the bearing housings. The small holes, drilled in the crankcase above the joints as shown, communicate with recesses machined around the bolts. These holes are made to prevent the oil from being forced upwards out of the engine through the bolt holes in the crank chamber.

Water Pump and Water-Cooling System.—Details of the construction of the water pump are shown in the sectional drawing, Fig. 31. The body of the water pump is attached to the top face of the separate aluminum casting which forms the cover of the magneto driving gears. This gear cover is attached by bolts and studs to the rear end of both the top and bottom halves of the crank chamber. As may be seen in the general arrangement drawing of the engine, the water-pump spindle is driven through a dog clutch at its lower end by a short vertical spindle running in a bronze bush bearing; this spindle is driven by a bevel gear meshing with the main bevel fixed on the rear end of the crankshaft. The water pump vertical spindle, which is 20 mm. in diameter, runs in a bronze bearing, which is lubricated by a screw grease lubricator. The top portion of the pump spindle bearing is cupped to form the housing for a thrust ball race, above which is fixed the pump rotor. The pump rotor is a gun-metal casting, having six helical vanes, 110 mm. in diameter. The rotor is fixed to the pump spindle on a taper and key, and is locked by a domed nut and locking washer.

The lower half of the pump body is an aluminum casting with an inlet passage 64 mm. in diameter, to which the diagonal water pipe from the radiator is coupled by a rubber connection. The top half of the water pump body, which is a gun-metal casting, is formed with six helical passages leading in a reverse helical direction to the water-pump rotor. These passages connect with the common vertical outlet passage in the top of the body casting, as shown in Fig. 31. The outlet diameter is 50 mm.

The center portion of the top body casting inside the helical passages above the pump rotor is domed and fitted with a screwed plug, 38 mm. in diameter. This plug is drilled with a 3.5 mm. hole, evidently to prevent an air lock. Two 3-mm. holes are also drilled in the bottom of the rotor between the vanes for the same purpose. It is noticeable that the steel ball-thrust race is working to all intents in the water flow, and must, therefore, suffer by corrosion.

Water Circulation.—The circulation of the water-cooling system is clearly set out in the diagrammatic drawing, Fig. 26, from which it will be seen that the water pump delivers directly into the bottom of the water jacket which surrounds the body of the rear carbureter. As already mentioned, the two carbureters are attached by long studs to the ends of the front and rear cylinders, and the water, after passing through the rear carbureter water jacket in a special passage round the throttle valve, enters the rear cylinder through the two circular water-joint flanges. The water passes through the passages cast in each of the six cylinder heads to the front carbureter or water jacket, whence the water is forced out through a vertical passage 50 mm. diameter through a diagonal pipe leading to the radiator, which is slung above the engine from the center section of the top plane.

The radiator is of entirely new design, and is shown in Fig. 33, fitted to a 260-h.p. Mercedes engine. According to a French report, this radiator is very efficient. It is of semi-circular shape in front elevation, and is formed of hexagonal honeycombed tubes, and exceptionally deep in cross-section, measuring 22 cm. The radius of the semi-circle in elevation is approximately 32 cm. The radiator is divided into two separate sections by a vertical baffle plate, as shown in the diagram, Fig. 26. Water enters the bottom of the left-hand side of the radiator through the pipe leading from the front carbureter, and flows upwards through the tubes of the left hand of the radiator to the top portion of the tank which completely surrounds the radiator. The water passes over the top of the dividing baffle plate, through the right-hand section to the bottom of the radiator, and out at the rear side, which is connected with a 50 mm. pipe to the inlet passage of the water pump. The front of the radiator is fitted with a blind, which is operated by a cable from the pilot's seat.

Ignition.—The two Bosch Z.H.6-type magnetos are mounted on brackets cast on the rear end of the base chamber, and are driven at 1.5:1 engine-speed directly off the rear end of the camshafts. The ignition point is set 38 degrees early. Two Bosch 3-point plugs are somewhat inaccessibly fitted in the head of each cylinder between the twin water connections as previously described. The plug bosses are water cooled. The high-tension cables are carried in fiber tubes bolted to the cylinders, and the leads to the plugs are taken through fiber blocks mounted on the sheet-metal brackets on the cylinder heads. The order of firing is: 1-5-3-6-2-4.

Carbureters.—The two carbureters are attached to the heads of the front and rear cylinders, as already described, and their general construction is shown in the cross-sectional drawing, Fig. 32. The carbureters follow the distinctive principles of the well-known Maybach design, but are slightly modified in many of their details as compared with the earlier engines. These carbureters are designed to be used in conjunction with a petrol-pump system, and in place of a float chamber employ two small constant-level overflow tanks inside each carbureter. This method obviates the necessity for using pressure-release valves in the petrol system. Referring to the sectional drawing of the carbureter, Fig. 32, they are constructed of a cast aluminum water-jacketed body, in the center of which is mounted the rotary barrel-type throttle, open on one side to the curved passage to the induction pipe, and on the other to the mixing chamber above the jets. The main air supply is taken through a passage cast in the aluminum carbureter body directly below the throttle and the extra air supply is automatically controlled by a guillotine-type shutter, which works verti-

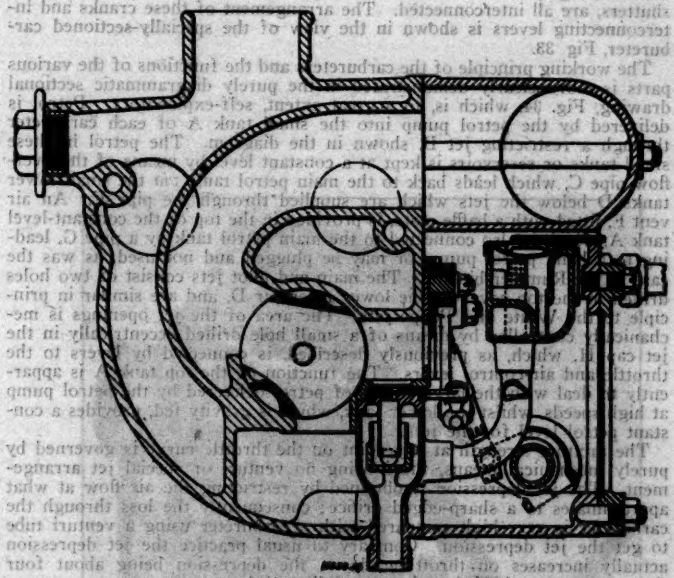


FIG. 32. SECTIONAL ARRANGEMENT OF CARBURETTOR.

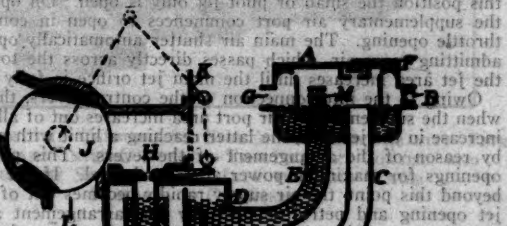


FIG. 34. DIAGRAMMATIC SECTION OF CARBURETTOR.

cally just in front of the jet-control lever. The throttles of both carbureters, together with the mechanically-controlled jets and the extra air shutters, are all interconnected. The arrangement of these cranks and interconnecting levers is shown in the view of the specially-sectioned carbureter, Fig. 33.

The working principle of the carbureters and the functions of the various parts is more clearly demonstrated in the purely diagrammatic sectional drawing, Fig. 34, which is, to a great extent, self-explanatory. Petrol is delivered by the petrol pump into the small tank A of each carbureter through a restricting jet B, shown in the diagram. The petrol in these small tanks or reservoirs is kept at a constant level by means of the overflow pipe C, which leads back to the main petrol tank, *via* the small lower tanks D below the jets which are supplied through the pipe E. An air vent F, fitted with a baffle plate, is provided in the top of the constant-level tank A, which is also connected to the main petrol tank by a pipe G, leading to a hand petrol pump, or may be plugged and not used, as was the case in the Rumpler biplane. The main and pilot jets consist of two holes drilled in the top cover of the lower reservoir D, and are similar in principle to the White and Poppe jets. The area of the jet openings is mechanically controlled by means of a small hole drilled eccentrically in the jet cap H, which, as previously described, is connected by levers to the throttle and air-control levers. The function of the top tank A is apparently to deal with the large excess of petrol delivered by the petrol pump at high speeds, whilst the lower tank, which is gravity fed, provides a constant petrol level for the jets.

The mixture strength at any point on the throttle curve is governed by purely mechanical means, there being no venturi or special jet arrangement. The jet depression is obtained by restricting the air flow at what approximates to a sharp-edged orifice; consequently the loss through the carbureter is very high compared with a carbureter using a venturi tube to get the jet depression. Contrary to usual practice the jet depression actually increases on throttling down, the depression being about four times as great at half-throttle as at full throttle.

Control Adjustment.—When running slowly the throttle is slightly open, the supplementary air port closed, and the air shutter practically closed; in this position the small or pilot jet only is open. On opening the throttle, the supplementary air port commences to open in conjunction with the throttle opening. The main air shutter automatically opens in proportion, admitting more air which passes directly across the top of the jets, and the jet area increases until the main jet orifice is fully open.

Owing to the interconnection of the control levers, there comes a point when the supplementary air port area increases out of all proportion to the increase in the jet area, the latter reaching a limit with no further increase by reason of the arrangement of the levers. This point represents the openings for maximum power at ground level. If the throttle is opened beyond this point, the air supply rapidly becomes out of proportion to the jet opening and petrol supply. By this arrangement a simple form of altitude control is provided. The various positions for "slow running," "slow speed," "full speed," and "altitude" are marked on the body of the carbureter, and are indicated by a pointer attached to the throttle lever of the rear-end carbureter. By means of these markings on the carbureter the correct settings of the throttle lever can be checked when the engine is installed in the machine. The translation of these markings on the carbureter is as follows:

LE	= Leer	= Running Light.
LA	= Langsam	= Slow Speed.
V	= Voll	= Full Speed.
H	= Hohe	= Altitude.

FIG. 33.—ONE OF THE CARBURETERS BROKEN TO SHOW GRAVITY PETROL RESERVOIRS AND INTER-CONNECTED THROTTLE, JET CONTROL AND GUILLOTINE SHUTTER; ALSO INDUCTION AND WATER PASSAGES TO CYLINDERS.

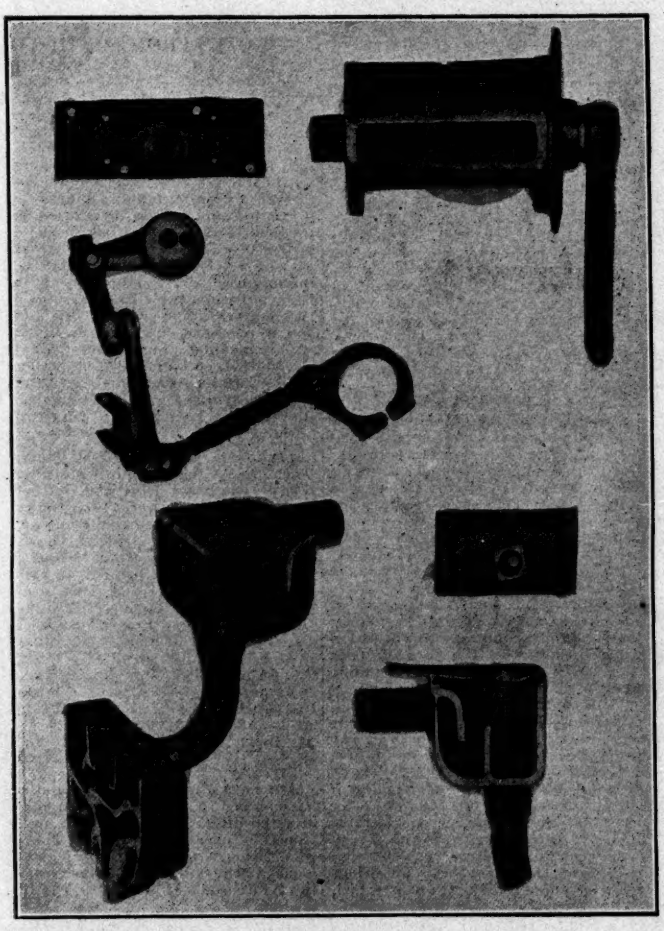


FIG. 37.—THE THROTTLE AND JETS DISMANTLED; AIR SHUTTER; AND SECTIONAL VIEWS OF PETROL RESERVOIRS IN CARBURETERS.

The induction pipes are aluminum castings, and are of exceptionally neat design, being, as shown in Fig. 35, joined together in one straight pipe. A cast-aluminum baffle plate drilled in the center with a 20-mm. hole is provided in the center of the two induction pipes between the front and rear three cylinders. This baffle is evidently fitted to equalize the gas supply to all cylinders. The design of this equalizing baffle is shown in the

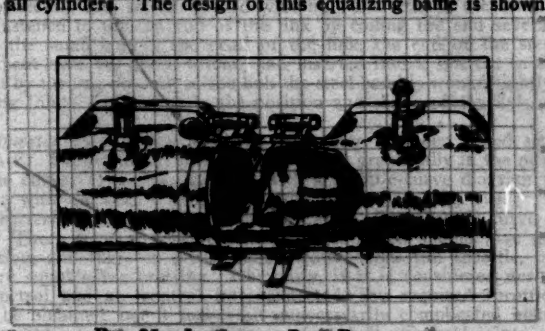


FIG. 35. INDUCTION PIPE BAFFLE AND PRIMING CUPS.

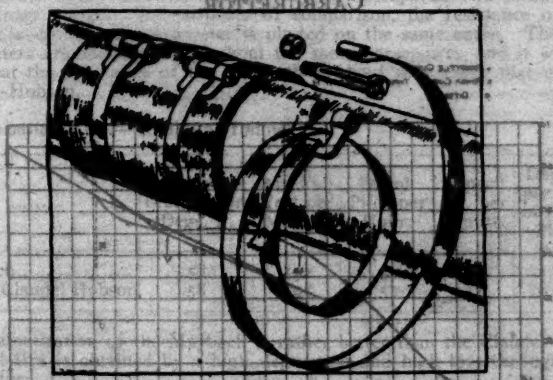


FIG. 36. STEEL BAND JOINT CLIP USED ON RUBBER CONNECTIONS OF INDUCTION PIPE AND WATER JOINTS.

perspective sketch, Fig. 35, which also shows the method of joining the ends of the induction pipes by rubber connections and band clips. Details of the six small primer valves fitted into the top of the induction pipe are also shown in the sketch. Fig. 36 clearly shows the construction of the interesting type of band clip which is used for making the connections in the induction pipes and also for most of the water joints on these engines. These clips are now extensively used on all enemy engines. The clips

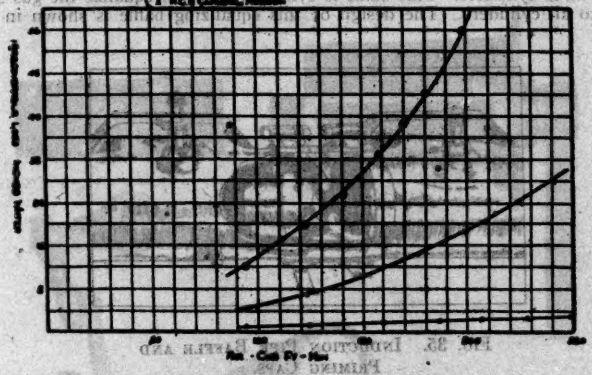


FIG. 39. DIAGRAM OF COMPARATIVE TESTS OF CARBURETTOR.

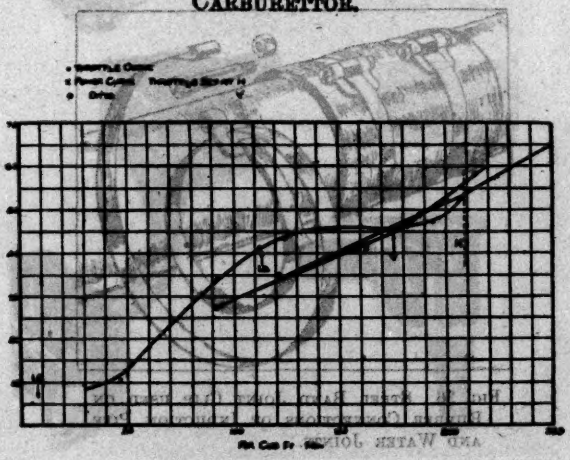


FIG. 38. CARBURETTOR TEST; THROTTLE CURVE DIAGRAM.

are made of thin strips of annealed spring steel; these are wound twice round the rubber rings and fastened by small steel links and bolts. In starting to make this joint, the end of the steel band is first bent over the bottom bar of the connecting link, and then wound twice round the rubber ring, the free end of the steel band passing through the center portion of the connecting link on the second winding. The end of the steel band is then fastened to the top end of the link by coiling the end several times round the small bolt in the head of the link, the end of the steel band being bent over for about 2 mm., and inserted into a slot cut in the bolt, so that by turning the bolt in the link the steel band is wound tightly round the ring joints, and owing to the position and shape of the link, remains fixed.

Carbureter Tests.—The Maybach carbureter has been tested separately at R.A.E. The results of these tests is shown graphically in the power and throttle curves (Fig. 38), together with a throttle-curve diagram in comparison with the H.C. 8 Claudel-Hobson carbureter (Fig. 39).

Characteristic Curves.—(*a*) Throttle Curve.—The throttle curve appears good. There is an excess of petrol at first which should give good acceleration, and there is a fair range, down from full throttle, of weak mixture suitable for cruising. Obviously, if it is considered desirable to have petrol and air control interconnected, any required throttle curve could be obtained, but it makes an unnecessarily complicated instrument. In this particular carbureter there are four ball universal joints and two sliding blocks.

(*b*) Power Curves.—The power curves are satisfactory, the mixture being practically constant over a large air speed range.

Resistance Test.—The resistance is very high compared with carbureters of ordinary design. For purposes of comparison, the resistance of the H.C.8 Claudel-Hobson carbureter is plotted on the same curve. The two carbureters are for engines of about the same horsepower, and it will be seen that the resistance of the Maybach is about nine times that of the Claudel-Hobson.

Atomization Test.—Results as follows:

.....	Air Temperature Degs. F.	Air, Cub. Feet per Minute.	Petrol, Pints per Hour.	Deposit c.c. per Minute.	Deposit per cent of Petrol Flow.
Maybach.....	58	192	50	23	4.85
H. C. 8 Claudel Hobson	58	195	51	32	6.62

The atomization of the Maybach is rather better than the Claudel-Hobson, but this is obtained by a high depression and high loss through the carbureter.

Conclusion.—The carbureter seems satisfactory on throttle and power curves and atomization, but the resistance is very high and the weight excessive. The throttle curve is obtained by a complicated mechanical system which would need careful first adjustment and constant adjustment for wear. The air is taken in through the moving parts, and if any dust is present the mechanism is very liable to stick or to render the control very hard to operate.

Petrol-Supply System.—Petrol is supplied to the two carbureters by a small double-acting duplex petrol pump, which is attached to the rear end of the bottom half of the oil base, and is driven at half engine speed directly off the rear end of the main oil-pump spindle, the petrol-pump driving shaft being coupled to the oil-pump spindle by a dog clutch. The in-

stallation of the petrol pump is clearly shown in the general arrangement drawing of the engine, and the details of the petrol supply system are shown in the diagrammatic drawing (Fig. 26).

Regarding the installation of the petrol-supply system to the petrol pump in the machine, it is assumed that the main petrol tank is provided with

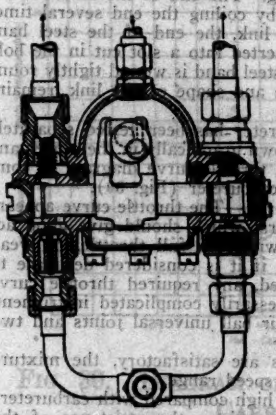


Fig. 40. Sectional Arrangement of Petrol Pump.

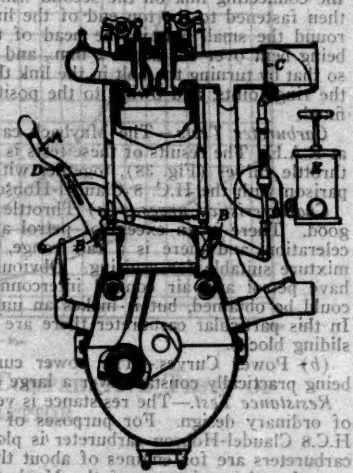


Fig. 42. Diagrammatic Section of Engine showing Functions of Starting Mechanism.

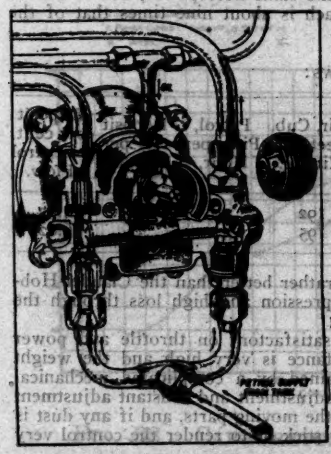


Fig. 41. Sectional Sketch of Petrol Pump.

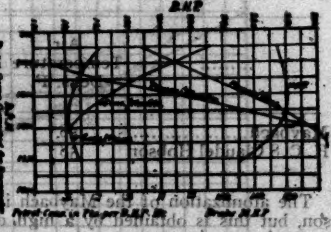


Fig. 43. Calibration Curves: R.A.E. Tests.

the ordinary float regulator, which has always been fitted inside the petrol tanks for regulating the supply through the petrol pumps; but owing to the fact that the machines from which these new Maybach engines were taken are completely destroyed no information regarding the construction

and installation of the petrol tanks is available. In the Rumpler machines fitted with the 260 h.p. Mercedes engines two petrol tanks are fitted. The main tank is in the center of the fuselage behind the pilot's seat, and the auxiliary tank is situated under the pilot's seat.

Petrol Pump.—The design of the compound petrol pump is shown in the sectional drawing, Fig. 40, and also in the sectional perspective sketch, Fig. 41. The pump consists of two opposed cylinders, in which a reciprocating plunger works, the ends of which operate as single-acting pumps. The pump plunger carries a yoke fitted with a sliding bush, which forms the crankpin bearing of the small pump crankshaft, driven, as already described, off the rear end of the oil-pump spindle at half engine speed. The bore of the petrol pump plunger is 15 mm. and the stroke is 17 mm. The outer ends of the pump barrels are as shown, and are fitted with screwed plugs, which form compression chambers.

The small non-return suction valves are situated directly above and below the two compression chambers, and the two petrol delivery valves are fitted above the compression chambers. The valves communicate with the compression chambers through small ports drilled in the pump barrel. Both the suction and delivery valves are of the poppet type, and each delivery valve is spring loaded by means of a small brass wire coil spring. The valves are supported in guides drilled in the unions which form the valve boxes, and to which the petrol-delivery pipes from the petrol tanks and to the carbureters are connected. The internal diameter of both these pipes is 10 mm. It will be noticed that both the suction valves are fed by the same petrol-supply pipe as shown in the sketch.

The center portion of the petrol-pump body, which is a gunmetal casting, forms a small circular crank chamber for the pump plunger, and is fitted at the bottom with a detachable cover plate, which is secured by six set screws. The whole of the pump-crank chamber is filled with oil under pressure from the main lubrication system, through an oil pipe connected to the center of the pump-body casting, and leading from the rear end of the detachable main oil-lead pipe on the engine. By this means the efficient lubrication of the yoke and sliding bush and also of the seating of the pump plunger is assured.

R.A.E. Tests of Petrol Pump.—Flow tests at three speeds with varying heads were taken; in order to represent working conditions in the engine, flow tests were also taken through jets similar to those employed on the engine.

Delivery Tests.—The petrol pump was run at three speeds, viz.: 550, 800, and 1,275 r.p.m., and the outlet pipe was connected to the pump against a variable head from zero to 6 pounds per square inch. The maximum and minimum petrol delivery at these speeds and heads are given in the following table:

Revs. per min.	Maximum flow. Zero head.	Minimum flow. 6 lbs. head.
550	300 pints per hour.	167 pints per hour.
800	422 pints per hour.	264 pints per hour.
1,275	630 pints per hour.	498 pints per hour.

Fig. 42 shows the graphs of these tests with varying heads plotted against delivery at constant speeds. These curves are quite normal, and it is evident from them that the valves are acting well, and that the drop in delivery with increasing head resistance is due to cavitation. The pressure of the oil in the pump case during these tests varied between 15 and 20 pounds per square inch, and the leakage of oil and petrol past the end bearing of the pump spindle during one hour's run was approximately one pint.

Tests Through Engine Delivery Jets.—In the engine the petrol is delivered through restricting jets (Fig. 34) into the constant-level tanks, which feed the jet chambers below, and are provided with overflows to the main

petrol tank. In the tests both outlets of the pump were connected by a Y piece with one delivery pipe, the end of which was closed by a plate 1/16 inch in thickness, having two 0.07-inch diameter jet holes drilled through. The petrol pump was run at the three speeds as in the former tests, and the following results recorded of delivery and head resistance:

R.P.M.	Head resistance.	Pints per hour.
550	4.5 pounds.	160 pints.
800	9 pounds.	218 pints.
1,275	23 pounds.	360 pints.

Starting Gear.—The principle of the Maybach starting gear is now so well known that only a brief description of the mechanism and a few details of the construction will be necessary. In the semi-diagrammatic cross-sectional drawing of the engine, Fig. 42, the working principles of this simple and distinctive starting mechanism are clearly shown. By the depression of the hand lever A on the induction side of the engine, all the tappets are lifted off their cams through the action of small lugs formed on the top of the tappets, which fit into slots cut in the tubular lay shafts BB. All the valves, both inlet and exhaust, are thus opened in the cylinder heads; and at the same time the hand lever A closes the shutter C in the exhaust manifold by the connection of levers shown in the drawing. The valves and exhaust shutter are then locked in this position by a peg, which is inserted in two holes D, which now coincide in the hand lever A. By the action of a large hand suction pump E in the pilot's seat, gas is then drawn into the combustion chambers through the inlet valves and induction pipe from the carbureters as indicated in the diagram. When the cylinders are charged the valves are returned to their normal positions by the withdrawal of the locking pin in the hand lever A, and simultaneously the free passage in the exhaust manifold is again opened by the shutter C. Ignition is then effected by means of a Bosch hand-starter magneto in the pilot's seat. It will be noticed that the exhaust shutter lever is provided with a spring-loaded tie rod to insure a true seating for the shutter. In order to prevent the engine from being started until the hand lever has been released to its off position, the locking hole in the lever which takes the pin is made in such a way that only a special form of locking pin can be used. For this purpose the handle of the starting magneto is made easily detachable, and is used for the purpose of locking the hand lever A. It is thus practically impossible to cause ignition with the valves open and the exhaust passage closed, which, of course, would cause firing back into the carbureters, and probably also result in destroying the hand pump.

ENGINE-TEST REPORT AT R. A. E. 5.4.18.

Calibration and Endurance Tests.—The Maybach engine (No. 1261), after several slight repairs had been carried out to the cylinders and propeller hub flange and coupling, was erected on a test bed, coupled to a Heenan and Froude dynamometer, and submitted to the following power and consumption tests, including a one-hour's duration test at normal speed. The results of the calibration tests are as follows:

R.P.M.	1,200	1,300	1,400	1,500
B.H.P.	258	279	294.5	304.5
Brake, M.E.P.	120.5	120.3	118	113.9
Petrol consumption in pints /B.H.P. hour	.53	.52	.526	.545

The results of these tests are shown graphically on the diagram (Fig. 43).

One-Hour Test.—At the conclusion of the above tests a run of one hour's duration at normal revolutions (1,400 r.p.m.) was carried out with the following results:

Average B.H.P.	290.
Petrol consumption	20 gallons = .55 pints per B.H.P. hour.
Oil consumption	11 pints = .038 pints per B.H.P. hour.
Oil pressure	5 lbs./sq. in.
Oil temperature	67°C.
Water temperature (inlet)	57°C.
Water temperature (outlet)	68°C.

Valve-Timing During Tests.

Inlet.	Exhaust.
O. 8°E.	O. 33°E.
C. 35°L.	C. 7°L.
Magneto advance	38°E.

Running was steady at all speeds between 900 and 1,400 r.p.m., but owing to the fact that the propeller hub flange on the crankshaft was damaged, and was running slightly out of truth, the vibration became excessive above 1,400 r.p.m. Considerable trouble was experienced with the water connection between the cylinders on the exhaust side. The running became unsteady below 900 r.p.m.

Distribution.—Owing to the exhaust manifold being fitted as part of the engine-starting gear, it was not possible to form an idea of the distribution. A diagram of the inlet and exhaust valve lift is shown graphically in Fig. 20.

F. G. C. Ap.D. (L.)
May, 1918.

J. G. Weir,
Lieut.-Colonel, Controller,
Technical Department.

GENERAL DATA.

Make of engine and rated H.P.	Maybach 300 h.p.
Type number	1,261.
Number and arrangement of cylinders	Six vertical.
Bore	165.0 mm. = 6.50 in.
Stroke	180.0 mm. = 7.09 in.
Stroke/Bore ratio	1.09 :1.
Area of one piston	213.825 sq. cm. = 33.2 sq. in.
Total piston area of engine	1282.95 sq. cm. = 199.2 sq. in.
Swept volume of one cylinder	3848.65 cu. cm. = 235.3 cu. in.
Total swept volume of engine	23093.1 cu. cm. = 14120.0 cu. in.
Clearance volume of one cylinder	778.9 cu. cm. = 47.64 cu. in.
Compression ratio	5.95 :1.
Normal B.H.P. and speed	294.0 B.H.P. at 1,400 r.p.m.
Maximum B.H.P. and speed	304.5 B.H.P. at 1,500 r.p.m.
Normal B.M.E.P.	117.7 lbs. per sq. in. at 1,400 r.p.m.
Maximum B.M.E.P.	120.5 lbs. per sq. in. at 1,200 r.p.m.
Piston speed	1654.0 ft. per min. at 1,400 r.p.m.
Mechanical efficiency (calculated)	86.0 per cent.
Indicated mean pressure (calculated)	137.0 lbs. sq. in.
Fuel consumption per B.H.P. hour	0.536 pint = 0.473 lbs.
Brake thermal efficiency	28.9 per cent.

Indicated thermal efficiency	33.6 per cent.
Air standard efficiency	51.0 per cent.
Relative efficiency	65.9 per cent.
Cu. in. of swept volume per B.H.P.	4.80 cu. in.
Sq. in. of piston area per B.H.P.	0.678 sq. in.
H.P. per cu. ft. of swept volume	360.0 B.H.P.
H.P. per sq. ft. of piston area	212.4 B.H.P.
Direction of rotation of crankshaft	Anti-clockwise (facing propeller).
Direction of rotation of propeller	Anti-clockwise (facing propeller).
Type of valve gear	Overhead valve rockers and push rods.
Type of starting gear	Maybach of special design.
Number of carbureters	Two Maybach.
Bore of main jets	Variable from 0.0 to 2.5 mm.
Bore of pilot jets	Variable from 0.0 to 1.1 mm.
Fuel consumption per hour	19.33 gallons.

Valve Areas and Gas Velocities.

Diameters—	
Induction pipe	62.0 mm. = 2.44 in.
Inlet port	45 × 67 mm. = 1.77 × 2.64 in.
Exhaust port	45 × 67 mm. = 1.77 × 2.64 in.
Exhaust branch pipes	66.0 mm. = 2.60 in. (approx.)
Cross-Sectional Areas—	
Induction pipe	29.26 sq. cm. = 4.67 sq. in.
Inlet port	30.15 sq. cm. = 4.67 sq. in.
Inlet valve (π dh.)	4.416 sq. in. (total).
Exhaust valve (π dh.)	4.366 sq. in (total).
Exhaust port	30.15 sq. cm. = 4.67 sq. in.
Exhaust branch pipes	34.11 sq. cm. = 5.31 sq. in.
Gas Vecocities—	
Induction pipe	196.1 ft. per sec.
Inlet port	196.1 ft. per sec.
Inlet valve	208.0 ft. per sec.
Exhaust valve	210.0 ft. per sec.
Exhaust port	196.1 ft. per sec.
Exhaust branch pipes	172.5 ft. per sec.

Inlet Valves (Two per Cylinder).

Outside diameter	54.0 mm. = 2.126 in.
Port diameter (in cylinder head)	48.0 mm. = 1.89 in.
Width of seating	3.5 mm. = 0.137 in.
Angle of seating	30°
Radius under valve head	20.00 mm. = 0.787 in.
Lift of valve	9.45 mm. = 0.372 in.
Diameter of stem	11.0 mm. = 0.433 in.
Over-all length of valve	136.5 mm. = 5.373 in.
Number of springs per valve	One.
Free length of spring	52.5 mm. = 2.066 in.
Length of spring in position (no lift)	39.5 mm. = 1.55 in.
Mean diameter of coils	51.0 mm. = 2.00 in.
Gage of wire	No. 6 B.W.G.
Ratio length of spring/lift of valve	4.17 :1.
Weight of valve complete with spring	0.843 lb.
Weight of spring bare	0.281 lb.
Inlet valve opens, deg. on crank	8° early.
Inlet valve closes, deg. on crank	35° late.
Period of induction	223°
Inlet tappet clearance	0.3 mm. = 0.012 in.

Exhaust Valves (Two per Cylinder).

Outside diameter	54.0 mm. = 2.126 in.
Port diameter (in cylinder head)	48.0 mm. = 1.89 in.
Width of seating	3.5 mm. = 0.137 in.
Angle of seating	30°
Radius under valve head	9.0 mm. = 0.354 in.
Lift of valve	9.34 mm. = 0.368 in.
Diameter of stem	11.0 mm. = 0.433 in.
Length of valve guide	80.0 mm. = 3.149 in.
Over-all length of valve	152.5 mm. = 6.00 in.
Number of springs per valve	One.
Free length of spring	52.5 mm. = 2.06 in.
Length of spring in position (no lift)	39.5 mm. = 1.55 in.
Mean diameter of coils	51.0 mm. = 2.00 in.
Gage of wire	No. 6 B.W.G.
Ratio. Length of spring/lift of valve	4.21 :1.
Weight of valve complete with spring	0.881 lb.
Weight of spring bare	0.281 lb.
Exhaust valve opens, deg. on crank	33° early.
Exhaust valve closes, deg. on crank	7° late.
Period of exhaust	220°
Exhaust tappet clearance	0.4 mm.

Inertia Forces, Bearing Loads, Etc.

Weight of piston, complete with rings and gudgeon pin	14.05 lbs.
Weight per sq. in. of piston area	0.4235 lbs.
Weight of connecting rod complete	8.93 lbs.
Weight reciprocating part of connecting rod	3.305 lb.
Total reciprocating weight per cylinder	17.355 lb.
Weight per sq. in. piston area	0.538 lbs.
Length of connecting rod (centers)	310.0 mm. = 12.20 in.
Ratio. Connecting rod/crank throw	3.445 :1.
Inertia, lbs. sq. in. piston area, top center	137.0 lbs. sq. in.
Inertia, lbs. sq. in. piston area, bottom center	75.5 lbs. sq. in.
Inertia, lbs. sq. in. piston area, mean	53.25 lbs. sq. in.
Weight of rotating mass of connecting rod	5.625 lbs.
Total centrifugal pressure	1106 lbs.
Centrifugal pressure, lbs. sq. in. piston area	34.4 lbs. sq. in.
Mean average fluid pressure, including compression	48.0 lbs. sq. in.
Mean average loading on crankpin-bearing total from all source in terms of lbs. sq. in. piston area	118.0 lbs. sq. in.
Diameter of crankpin	66.0 mm. = 2.598 in.
Rubbing velocity	15.85 ft. sec.
Effective projected area of big end bearing	43.23 sq. cm. = 6.70 sq. in.
Ratio. Piston area/protected area of big end bearing	4.96 :1.
Mean average loading on big end bearing	585 lbs. sq. in.
Load factor on big end bearing	9270 lbs. ft. sec.

Cylinders.

Over-all height of bare cylinder from top of base chamber	479.5 mm. = 18.87 in.
Depth of spigot at base of cylinder	3.5 mm. = 0.13 in.
Diameter of cylinder over water jacket	185.0 mm. = 7.28 in.
Valve centers (between inlet and exhaust)	63.0 mm. = 2.48 in.
Thickness of flange at base of cylinders	12.0 mm. = 0.47.
Number of holding-down studs per cylinder	Four.
Diameter of holding-down studs	19.0 mm. = 0.74 in.
Thickness of water jacket	1.0 mm. = 0.039 in.
Mean thickness of combustion chamber wall	8.0 mm. = 0.31 in.
Mean thickness of cylinder barrel	3.0 mm. = 0.11 in.
Tensile stress	6640 lbs. sq. in. (approx.) (Assumed maximum pressure 450 lbs. sq. in.)

Piston.

Type of piston	Cast-iron (flat crown).
Diameter at top	164.25 mm. = 6.466 in.
Diameter at bottom	164.75 mm. = 6.486 in.
Length	511.00 mm. = 5.944 in.
Ratio. Piston length/cylinder bore	.914 :1.
Number of rings per piston	Three piston rings, one scraper ring.
Position of rings	All above gudgeon pin.
Width of rings	6.5 mm. = 0.255 in.
Gap of rings in cylinder	1.39 mm. = 0.055 in.

Connecting Rod.

Length between centers	310.0 mm. = 12.205 in.
Ratio. Connecting rod/crank throw	3.44 :1.
Little end bearing type	Floating cast-iron bush.
Floating bush, diameter, inside	38.0 mm. = 1.496 in.
Floating bush, diameter, outside	44.3 mm. = 1.743 in.
Floating bush, effective length inside	93.0 mm. = 3.661 in.
Floating bush, projected area of bearing on gudgeon pin	35.35 sq. cm. = 5.48 sq. in.
Ratio. Piston area/projected area of little end bearing	6.06 :1.
Big end bearing. Type	Bronze shell-lined white metal.
Big end bearing. Diameter	66.0 mm. = 2.598 in.
Big end bearing. Length (actual)	73.56 mm. = 2.893 in.
Big end bearing. Projected area	43.23 sq. cm. = 6.700 sq. in.
Ratio. Piston area/projected area of big end bearing	4.96 :1
Number of big end bolts	Four.
Full diameter of bolts	14.0 mm. = 0.551 in.
Diameter at bottom of threads	12.0 mm. = 0.472 in.
Total cross-sectional area, bottom of threads	4.520 sq. cm. = 0.70 sq. in.
Pitch of threads	1.5 mm.
Total load on bolts at 1,400 r.p.m.	5,824 lbs.

Total load on bolts at 1,600 r.p.m..	7,602 lbs.
Stress per sq. in. at 1,400 r.p.m.....	8,320 lbs. sq. in.
Stress per sq. in. at 1,600 r.p.m.....	10,860 lbs. in sq. in.

Crankshaft.

Number and type of main bearings..	Seven bronze shell-lined white metal.
Cylinder centers	187.0 mm. = 7.362 in.
Crankpins—	
Outside diameter	66.0 mm. = 2.598 in.
Inside diameter	38.0 mm. = 1.496 in.
Length	74.0 mm. = 2.913 in.
Journals—	
Outside diameter	66.0 mm. = 2.598 in.
Inside diameter	36.0 mm. = 1.417 in.
Length, propeller end	67.0 mm. = 2.638 in.
Length, rear end	67.0 mm. = 2.638 in.
Length, center	67.0 mm. = 2.638 in.
Length, intermediate	67.0 mm. = 2.638 in.
Crank Webs—	
Width	95.0 mm. = 3.740 in.
Thickness	23.0 mm. = 0.906 in.
Radius at ends of journals and crankpins	4.5 mm. = 0.171 in.
Weight of complete shaft	99.9 lbs.

Working Clearances.

Piston clearance, top (total).....	0.75 mm. = 0.029 in.
Piston clearance, bottom (total)....	0.25 mm. = 0.009 in.
Side clearance of connecting rod in piston (total).....	11.8 mm. = 0.464 in.
Side clearance of big end on crankpin (total).....	.44 mm. = .0173 in.
End clearance of crankshaft in main bearings	3.0 mm. = 0.118 in.
Clearance of valve stem in guide (inlet)	0.12 mm. = .00472 in.
Clearance of valve stem in guide (exhaust).....	0.15 mm. = .0059 in.

Lubrication System.

Number and type of oil pumps.....	Three, rotary gear.
Oil consumption per hour	11.0 pints.
Oil consumption per B.H.P. hour...	0.037 pints.
Oil temperature	65 deg. Cent.
Oil pressure	5.0 lbs. per sq. in.
Specific gravity of oil	0.899 s.p.g.
Ratio. Pump speed/crankshaft speed	1:2.
Pump delivery (calculated at 100 per cent volumetric efficiency).....	91 gallons per hour at normal engine revs.

Ignition.

Number and type of magnetos.....	Two Bosch.
Firing sequence of engine	1-5-3-6-2-4.
Ignition timing (fully advanced)....	38 deg. early.
Number of plugs per cylinder.....	Two.
Type of plugs	Bosch 3-point.
Ratio. Magneto speed/engine speed	1.5:1.

Metallurgical Analysis of Principal Parts.

	Carbon.	Graphitic Carbon.	Combined Carbon.	Silicon.	Manganese.	Sulphur.	Phosphorus.	Nickel.	Chromium.
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Cylinder head	—	2·57	0·21	1·69	0·70	0·121	0·241	—	—
Cylinder barrel	0·49	—	—	0·33	1·01	0·062	0·054	—	—
Cylinder water jacket	0·25	—	—	0·37	0·62	0·052	0·057	—	—
Piston	—	2·42	0·83	1·29	0·82	0·211	0·30	—	—
Gudgeon pin floating bush	—	2·50	0·66	1·43	0·76	0·146	0·47	—	—
Gudgeon pin	0·28	—	—	0·23	0·43	0·025	0·052	—	—
Connecting-rod	0·18	—	—	0·33	0·71	0·057	1·018	1·82	0·49
Inlet valve	0·49	—	—	0·20	0·48	0·061	0·041	4·01	0·51
Exhaust valve	0·40	—	—	0·20	0·36	0·019	0·025	2·62	1·16
Crankshaft	0·31	—	—	0·31	0·63	0·060	0·015	4·01	0·83
Camshaft	0·40	—	—	0·23	0·23	0·022	0·040	0·86	0·60
Camcore	0·18	—	—	0·23	0·75	0·039	0·014	0·47	—

	Lead.	Iron.	Tin.	Copper.	Zinc.	Antimony.	Silicon.	Manganese.
	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.	Per cent.
Big-end bearing metal	9·81	—	80·66	6·34	—	0·89	—	—
Main journal bearing	0·92	—	77·65	10·09	—	11·16	—	—
Crankcase	—	0·50	—	0·67	11·01	Aluminium (by Diff.) 99·57	0·65	Trace.

Mechanical Tests.

Mechanical tests were made on the crankcase and crankshaft, the results of which are given below :

Crankcase.

Mark.	Diameter.	Maximum Stress.	Elongation, per Cent.
T	0·253	11·05	—
L	0·254	11·25	—

Crankshaft.

Position.	Mark.	Diameter.	Yield Stress.	Maximum Stress.	Per cent. Elongation on 4A.	R. of A.	Impact.
		in.	Tons/sq. in.	Tons/sq. in.		Per cent.	Ft.-lbs.
End	1	0·254	59·3	68·1	14·8	50·8	17·21
Journal	2	0·254	—	—	—	—	—
Web	3	0·254	61·6	84·1	13·1	34·0	16·121
Long.	4	—	—	—	—	—	—
Crankpin, long	5	0·254	62·6	87·1	15·1	35·7	13·12
Web, trans.	6	0·254	61·4	85·3	7·2	16·5	6·5

GENERAL ANALYSIS OF WEIGHTS.

Description of Part.	No. per Set.	Average Unit Wt. Lbs.	Weight of Complete Set. Lbs.	Percentage Total Weight.
Cylinders, bare	6	32·75	196·50	21·50
Pistons, complete with rings and gudgeon pin set screws	6	12·20	73·20	8·11
Gudgeon pins	6	1·75	10·50	1·15
Connecting rods and floating bushes	6	8·08	49·12	5·39
Crankshaft, complete with oilrings	1	99·90	99·90	10·96
Crankshaft, complete with oil pin	1	4·00	4·00	0·44
Inlet valves	12	0·48	5·75	0·63
Exhaust valves	12	0·47	5·70	0·62
Inlet and exhaust valve springs	24	0·28	6·74	0·74
Inlet and exhaust valve collets, with cotters and locking device	24	0·12	3·00	0·32
Thrust, complete with ball races, propeller hub flange, and camshaft driving sprocket	1	17·63	17·63	1·94
Camshafts	2	10·00	20·00	2·20
Overhead valve rockers, complete	12	1·06	12·69	1·39
Overhead rocker bearings, complete	24	0·43	10·50	1·15
Valve tappets and guides	12	0·96	11·25	1·23
Crankcase, top half	1	24·30	24·30	10·66
Crankcase, bottom half	1	41·82	41·82	4·54
Bearing caps	6	2·69	16·12	1·77
Front bearing cap	1	5·60	5·56	0·61
Crankcase holding-down bolts, with clamps, nuts and washers	14	2·40	33·60	3·70
Induction pipe, complete	1	9·00	9·00	1·00
Propeller hub, with bolts and nuts	1	21·09	21·00	2·30
Inlet valve push rods	6	0·48	2·90	0·32
Exhaust valve push rods	6	0·48	2·87	0·33
Inlet and exhaust stiffening plates	2	2·00	4·00	0·44
Oil pumps, with driving gear	1	9·00	9·00	0·98
Rear cover plate	1	6·06	6·06	0·66
Front cover plate	1	4·00	4·00	0·44
Water pump, complete	1	8·50	8·50	0·93
Camshaft, oil and water pumps & driving gears	1	13·65	13·65	1·50
Wireless clutch	1	5·06	5·06	0·55
Revolution counter gear	1	2·62	2·62	0·28
Machine gun interrupter gear	1	1·82	1·82	0·20
Petrol pump	1	3·50	3·50	0·38
Magneto	2	10·75	21·50	2·36
Magneto wiring	1	4·75	4·75	0·52
Oil pipes	1	4·00	4·00	0·44
Self-starter gear	1	6·00	6·00	0·66
Exhaust manifold	1	27·00	27·00	2·96
Carburettors	2	16·00	32·00	3·51
Miscellaneous	...	...	3·75	0·41
Total weight of engine	...	...	911·00	100·00

Cooling System.

Number and type of water pumps....	One centrifugal.
Diameter of inlet pipe	54.0 mm. = 2.126 in.
Diameter of outlet pipe	50.0 mm. = 1.966 in.
Diameter of rotor	111.0 mm. = 4.46 in.
Water capacity of one cylinder....	1284.0 cub. cm.
Number and type of radiators	One, semi-circular honeycomb.
Ratio. Water pump speed/engine speed	2:1.
Water temperature, inlet	57 deg. C.
Water temperature, outlet	68 deg. C.

Petrol Pump.

Number and type of petrol pumps..	One Maybach, double-acting.
Bore	15.0 mm. = 0.59 in.
Stroke	17 mm. = 0.66 in.
Normal delivery	384 pints per hour at 800 r.p.m.
Maximum delivery	630 pints per hour at 1,275 r.p.m.
Ratio. Pump speed/crankshaft speed	1:2.

Weights.

Weight of complete engine, dry, with propeller boss and exhaust manifold	911 lbs.
Weight per B.H.P., ditto	3.10 lbs.
Weight of fuel per hour	139 lbs.
Weight of oil per hour (s.p.g. .899)..	12.36 lbs.
Total weight of fuel and oil per hour	151.36 lbs.
Gross weight of engine in running order, less fuel and oil (cooling system at .65 lbs. per B.H.P.)	1103.0 lbs.
Weight per B.H.P., ditto	3.79 lbs.
Gross weight of engine in running order, with fuel and oil for six hours (tankage at 10 per cent weight of fuel and oil).	2100.9 lbs.
Weight per B.H.P., ditto	7.14 lbs.

—"Engineering."

ELECTRIC ARC WELDING.*

By Robert E. Kinkead.†

In presenting a paper to a body of engineers on the subject of electric welding, it seems advisable to treat the subject in general terms and to cover, so far as possible, the whole field, allowing points of particular interest to come up in the discussion to follow.

There are two general kinds of welding, namely, that which requires heat and mechanical pressure and that which requires heat only. The first kind of welding includes forge welding, electric butt and spot welding and electro-percussive welding. The second kind of welding includes the thermit welding process, the oxy-acetylene and electric arc processes. The latter two processes are called autogenous welding processes. An autogenous weld is made by heating the metals to a liquid state and pouring

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them together into a homogeneous molten mass and allowing them to cool. The source of the heat required to melt the metal does not alter the principle of the operation, but does necessitate different methods of performing it. With very few exceptions the same operations may be accomplished with either the oxy-acetylene or electric arc process. The electric arc has a higher temperature than the oxy-acetylene flame and the great part of its heat is produced in an extremely localized area *within* the metal to be welded rather than external to it, as is the case in the oxy-acetylene flame. The heat of the arc being produced in the metal is more efficiently used so that it is possible to do at least three times as much welding with a given amount of heat produced by the electric arc as would be possible with the same amount of heat produced by the gas process. The approximate heat quantities involved in the welding arc are easily calculated from this formula

$$H \text{ equals } \frac{E \times I}{1{,}000} \times T \times 3{,}413.$$

H equals heat in British thermal units.

E equals average voltage across arc.

I equals average current in the arc circuit.

T equals time arc is in operation in hours.

The factor 3,413 is the heat equivalent of one k.w.-hour energy in British thermal units.

The heat liberated by the oxy-acetylene flame may be calculated knowing that a 1 cubic foot of an oxy-acetylene mixture gives about 1,550 British thermal units of heat.

The cost of producing a unit of heat for autogenous welding with the present prices of gas and electric power is about in a ratio of gas, 3; electric, 1. With an "effectiveness" factor of about 1 to 3 in the use of the two processes, it is evident that the cost of gas is in the neighborhood of nine times the cost of electric power doing the same work.

Up to the present time the electric-arc welding process has been used almost exclusively in the welding of steel. The very great localization of heat of the arc gives it a marked advantage over the oxy-acetylene plant for welding on boiler plate and sheet metal. The difficulties arising from expansion and contraction with the attendant buckling of the plate and cracking of the weld encountered in the use of the gas process are, to a certain extent, eliminated in the arc process due to this localization of the heat.

Gray iron may be welded with the carbon arc welding process, but the operation is somewhat more difficult than welding with the gas flame and perhaps requires more skill on the part of the operator.

The commercial applications of the electric-arc welding process are very interesting. By the use of the process it has been possible to accomplish certain things which were not possible before. It is often possible, by its use, to accomplish in a few hours essentially the same results which a few years ago would have required ten times the labor and perhaps ten times the actual working time. The use of electric power for autogenous welding is another triumph for electricity.

One of the most interesting applications of the electric-arc welding process is in the locomotive shops and round houses of the country. This is an application which is growing in importance daily. The steam locomotive is merely a large number of pieces of steel bolted or riveted together. Each piece and the whole is subjected to the most severe stresses. Most of the parts are subjected to excessive wear, and the boiler is worked to the limit of its capacity under most unfavorable conditions. Under present war-time conditions of railroad traffic motive power is being kept on the road double the number of miles that was considered possible two years ago, before finally coming back to the locomotive shop.

The electric-arc welding process is used to weld the flues into the back-flue-sheet of the boiler; to weld cracks and patches in the firebox; to weld in new sections of the flue sheet, door sheet or side sheets. Broken frames are welded in from six to ten hours without dismantling the engine. Worn links, guides, hangers, etc., are built up with new metal, re-machined and put into service in a few hours time. Worn flanges on the drive wheels are built up without dropping the wheels and the engine is put in service without turning the tyre.

I submit copy of a paper presented by E. Wanamaker, electrical engineer of the Rock Island lines, in which he presents some very interesting facts in this connection. One of the most interesting is that by the use of the welding process they have obtained from their present motive power equipment the same service that they would have obtained with 23 additional engines, but without the welding equipment. Mr. Wanamaker's paper on "War and Welding" is one which every railroad man in this country ought to read. If there is anyone who has any doubt as to what the engineers of this country must do as their share to help win the war, a study of the results which Mr. Wanamaker has obtained is to be recommended.

(*Copy of Paper.*)

WAR AND WELDING.

By Mr. E. Wanamaker, Electrical Engineer, Rock Island Lines.

In addressing the Western Railway Club on the subject of operating economy, I feel that it is entirely unnecessary to enumerate the reasons why it is imperative for the railway systems of the country to exert the maximum effort to move the freight of the country with the greatest possible dispatch. That is a lesson which we have learned as the result of the campaigns which have already been fought in Europe. We have the example of the remarkable effectiveness in transportation which has been reached by Germany, and, on the other hand, have witnessed the most complete failure of the Russian military effort, due to lack of proper transportation facilities. The paper presented to this club on September 18 by Mr. W. L. Park, vice-president of the Illinois Central Railway Company, together with the discussion following it, deals with this subject in a very excellent manner.

So far as we can see at present, the railroads of the country will not be burdened as seriously, due to the necessity of transporting the armed forces for the nation, as they will be burdened with the transportation of food for the seaports, and the multitude of commodities which require transportation to be manufactured into munitions of war, and which later must be delivered to a seaport for ocean transportation to the theater of war. This latter transportation service is in no way different in quality from the ordinary industrial transportation service which the carriers must render in time of peace. However, there is an enormous difference in the quantity of service required. The effectiveness of the nation in the war is a function of the time required to put effective fighting units in the field and keep them properly supplied with food and the implements of war. The ability of the railroads to meet the demands upon their transportation facilities will be an important factor in the time required for the nation to become effective on the battle fronts.

This discussion deals with one way of getting more transportation service out of the railway equipment in existence at the present time. Efforts spent in this direction are as patriotic and necessary as can be imagined.

I do not wish to depreciate for a moment the patriotism of the man who puts on the uniform and risks his life "to make the world safe for democracy." I do insist, however, that the patriotism of the civilians, particularly of the mechanical engineers in railway service, must not stop with the more or less superficial demonstration of loyalty to the stars and stripes. Those of us who cannot go to the fighting front must use the best of our abilities to get service, "and still more service," to paraphrase Lloyd George's quotation, from the motive power and rolling stock of the nation's railway systems.

The growing possibilities of the welding processes in motive power and rolling stock maintenance have been a source of amazement to every railway man who has come in contact with the practice. From an engineering standpoint, the welding processes are particularly interesting. In the matter of cutting metal the improvement in practice brought about by the introduction of the oxygen torch is particularly great. Only a few years ago the shearing of plates was the only method used for cutting plates, etc., to the proper size for manufacture and repair purposes. It was slow, inaccurate and wasteful. The cutting of large holes and special shapes after the plate had become a part of a boiler or machine had to be accompanied with drill and chipping hammer. The cutting of rivet heads, bolts and nuts was an operation which took an unreasonable length of time and required an expenditure of labor all out of proportion with the results accomplished. In such operations the application of the oxygen cutting processes saved from 50 to 90 per cent of the time required to perform the operation. Yet I regret to state that in many of our locomotive shops and car yards the old practices are still followed, and our man power for this cause has not produced the maximum result.

The matter of fastening two pieces of metal together is one of the fundamental problems of mechanical engineering, since all machines are made up merely of so many pieces of metal fastened together. The rivet is almost as old as the hammer and anvil. The threaded bolt and nut, as compared with the rivet, are recent inventions. The welded fastening came down to us from antiquity. The welded fastening has always been looked upon as a stronger joint than the riveted or bolted fastening. The reason that it was never more widely used until recent years has been due to the fact that only one way of making a welded joint was known, and that this was not applicable, except to a limited range of joints. As a general proposition the riveted joint, or bolted joint, has a tensile strength which is considerably less than the tensile strength of the original piece, while the welded joint is as strong as the original section. For instance, two pieces of boiler plate may be riveted together with four rows of rivets and double straps, and the tensile strength of the joint will not exceed 85 per cent of the strength of the original plate, yet it is quite a simple matter to weld a joint in the same plate, either by the old forge welding processes or by later autogenous welding processes, and obtain a joint that is as high in tensile strength as the original plate. The reason more of such welded joints have not been produced prior to the introduction of the autogenous welding processes has been due entirely to the difficulty experienced in applying a forge weld to the structures made of plate. The welded joint, and particularly the butt-welded joint made by an autogenous welding process offers greater strength, with a thickness for practical purposes equal to the thickness of the original plate. This is a feat in mechanical engineering which has been unheralded, but ranks with the introduction of the power hammer, the pneumatic tool, and other great strides in mechanical engineering progress.

There was a time when worn surfaces of the steel parts in a machine made it necessary to scrap those parts. There was no method, aside from the using of chafing plates, of avoiding this practice. The wearing

of a fraction of a cubic inch of metal from one of the wearing surfaces of a steel part made it necessary to scrap the labor investment in that particular part entirely. It was only with the introduction of the autogenous welding processes that it became possible to stop this economic waste. This also ranks as an important engineering achievement, and it has only been in the last three years that the possibilities in this direction have begun to be realized. While it has not been possible to apply this salvage to cast-iron parts to any great extent, the tendency has been towards the introduction of steel castings in place of gray iron castings to accomplish the desired result.

The repair of broken members of a machine has been another important work of the autogenous welding process. Prior to their introduction certain steel parts could be welded in the forge fire, but the range of the work was very limited and in many cases its cost equal to the cost of a new part. Now, however, with the exception of cast-iron parts, the failure of which would bring death and destruction, practically all parts of a machine which break may be safely repaired.

Viewed from the perspective of the last 50 years, the achievements made possible by the autogenous welding processes in the last five years are probably the most remarkable of any in the mechanical field. It is my purpose to point out so far as is possible how far these new practices may be utilized on steam railways to enable the railway systems of the country to meet the present emergency in the matter of service demanded from motive power and rolling stock equipment. It seems advisable to preface my remarks on the newer processes with a short discussion of their relationship to the older and more established practices.

Forge welding has probably been practiced ever since man had intelligence enough to construct a crude anvil and a hammer. It rests on the fact that when two pieces of iron are raised to such temperature that they become plastic, they may be forged together and will become, at the junction, a homogeneous mass. The art of forging depends upon three things:

1. The surfaces which are to be welded together must be clean so that the molecules of iron may come into intimate contact during the operation.

2. The temperature of the metals must be such that cohesion of the molecules may take place, but must not be high enough to depreciate the physical properties of the metals.

3. The mechanical pressure must be applied in such a manner as to bring the molecules into such intimate contact that cohesion will take place.

The first of the new welding processes, which may properly be called autogenous processes, since the welding takes place more or less automatically, was the thermit welding process. In this process the metals to be welded, together with the additional quantity of metal of approximately the same composition, are heated to a liquid state by a chemical reaction, so that cohesion takes place while they are in the molten state, *without the mechanical pressure.* This process found a wide application in the repair of broken steel parts on the railroads and the economy which resulted from its use without doubt aggregated millions of dollars.

Welding with the oxygen and fuel-gas flame was the first widely used autogenous welding process. In this case the heat for welding is produced by the chemical union of oxygen and some fuel gas, such as acetylene. The gas flame as a tool in the railway shop is best described by the once fictitious term, "putting on tool." The parts to be welded are heated to a liquid state at their surfaces by the heat of a flame and the metal to be added is melted simultaneously. The union, or fusion, takes place and the metal in the joint cools off, leaving a homogeneous mass which we call a weld. Since every one is more or less familiar with the gas-welding process, this paper will deal primarily with the electric-welding

process and an extended discussion of the gas process will be omitted. It is sufficient to state that at the present moment the essential difference is in the method of producing the heat for welding, rather than in the fundamental principles involved.

In passing it may be stated that there is a form of electric welding which is not an autogenous method of welding, but which is increasing in prominence in railway work. I refer to butt or spot welding work. In these processes both heat and pressure are used to produce the weld. The heat is produced by the resistance offered to the passage of an electric current through the pieces to be welded. The temperature is raised to the point at which the metals become plastic and the pressure applied. In principle the processes are identical to the forge welding process. Butt welding as applied to the safe ending of flues offers an attractive saving and will undoubtedly become standard practice on all railways in the near future.

There are two kinds of electric arc welding, known respectively as carbon electrode welding and metal electrode welding. In the former an arc is drawn between a carbon electrode and the piece to be welded and the metal to be added fed into the arc in the form of a "melt bar." This process is not used extensively in railway work, due to the fact that welding may only be done in the horizontal plane in this manner and that the work is in general inferior to that which is possible with the metal electrode process.

The metal electrode process uses, as the name implies, a metal electrode—the arc being drawn between the electrode and the piece being welded. The heat of the arc melts the metal of the piece and the metal of the electrode simultaneously. As the metal of the electrode melts it is drawn across the arc to the molten metal of the piece where a complete and homogeneous union is formed, which we call an autogenous weld. I say the metal of the electrode is *drawn* across the arc rather than that it falls through the arc advisedly, since it will flow straight overhead as well as straight downward. The temperature of the arc is extremely high at its center, actually vaporizing the metal to form the visible arc. With the exception of work with certain electrodes (manganese steel and slag-covered electrodes), the electrode is always made the cathode or negative—that is, the current of electricity flows *from* the piece being welded to the metal electrode. The reason for this practice is that the greatest amount of heat in an electric arc is liberated at the point at which the current passes from the solid medium to the heated vapor of the arc. Since the metal of the piece has more mass, and conducts the heat away from the point at which the welding is being done more rapidly than the electrode, it is desirable to have the greatest amount of heat on the piece. Due to the composition of the manganese and slag-coated electrodes, it is necessary to make these electrodes the positive.

The characteristic of the wire which is used for the metal electrode is an important matter. A dead soft steel wire has been found entirely satisfactory for this work. Not all dead soft steel wire, however, works well in the arc. Uniform mechanical treatment in manufacture and careful annealing are required to secure the best results. This wire can be secured at the present time at a cost ranging from approximately 6 cents to 10 cents per pound. Good welding wire may be relied upon to give metal in the weld which has a tensile strength of from 50,000 to 55,000 pounds per square inch. The deposited metal is very soft and easily machined, although it is without appreciable elasticity. When properly deposited, the metal is practically free from blow holes and slag inclusions, although a certain amount of oxide in solution is unavoidable. Except for the absence of elasticity, the deposited metal is of the nature of cast steel.

The voltage required for metal electrode welding is approximately 20 volts and direct-current power is necessary. The various types of welding equipment are merely different plans for rendering available a rather heavy current at this voltage, and the power economy of the several systems for obtaining this result varies over a wide range. Since we have gone into the matter on a rather extensive scale on the Rock Island Lines, it may be of interest to analyze briefly the conditions which led us to decide on a certain type of equipment.

A little over five years ago, the Rock Island installed four electric welding units of the first type that had been manufactured in this country. During the time since their purchase, and up to the present, we have been quietly investigating and developing our electric welding practice. As a result, we found that to obtain a suitable welding system it was necessary that we secure a light, compact, portable unit, of few parts, and of extreme simplicity, which would be entirely free from complicated mechanical regulating devices.

In planning the installation, a careful analysis was made of the service which is demanded of electric welding equipment in railroad shops and engine houses, and an attempt was made to design an installation which would show under such conditions maximum reliability and flexibility with minimum installation and operating expense. It appeared certain that the welding process would find a wider field of application than was apparent at that time, and the installation was made in such a manner as to lend itself readily to enlargement should that become necessary. And I might say here that in view of our experience and results we were indeed fortunate in rendering an expansion of the welding facilities easy. The lack of standardization of operations in the practice at that time made it extremely difficult to predict accurately the size of installations which would be required at the various points on the system, so it appeared desirable to install equipment which could be moved from one point to another, until proper distribution could be obtained.

It was considered particularly desirable to have the arc welding equipment available at all points in the shops and engine houses, since it was quite evident that the advantage of the low cost of the arc welding process would be lost if, for instance, the locomotive had to be moved from a haphazard location to a point where the welding power would be available. These features led to an analysis of distributing systems for the welding power and showed the necessity of using portable arc welding equipment, similar, so far as possible, to the portable gas welding outfits. With the total capacity divided at each shop among several units, it appeared certain that so long as power was available, there would not be a complete shutdown of the welding equipment. Each operator would be entirely independent of the others, although as many as necessary could be concentrated on any engine or job in the shop.

The operating economy, while not a deciding factor, was important, since at some shops the power plants were loaded to their capacity, and at other points the power purchased from small central stations was rather dear. Under these conditions, the variable-voltage type of equipment was considered the best because this type eliminated the resistance ballast from the arc circuit, thus increasing the power economy of the units to such a degree that a single operator unit may be operated from a power line large enough to carry a 5-horsepower motor.

As a result of the analysis of the requirements of the shops and car repair yards, 33 individual unit welders were installed, 10 of which were mounted on brackets on columns in the largest locomotive shop, while the other 23 were portable machines weighing complete with truck, approximately, 1,700 pounds. All of these units are motor generators having inherent regulation and an arc stabilizer. The portable units are equipped with ball bearings which require lubrication about once in six months.

Part of the machines are equipped with 230-volt direct-current motors and part with 440-volt three-phase, 60-cycle motors, these two kinds of power being the standard on the Rock Island Lines.

I might say in passing that at the time we purchased the portable equipment the manufacturers of the welding outfit were only able to furnish a standard industrial truck for this purpose. We have since found from actual practice exactly what is required in this line and are developing a standard truck which meets the requirements much better than an ordinary industrial truck. One of the manufacturers will shortly offer this development on the market as standard equipment.

The portable unit system was found to be from 30 per cent to 40 per cent lower in total initial installed cost due to the elimination of the low voltage distributing system.

With a system such as has been installed on the Rock Island Lines, with portable welders, the installation of electric welding equipment is converted into a system proposition rather than a series of plants to take care of certain shops or terminals. The plan is extraordinarily flexible and has many desirable features that would be impossible to obtain with any other plan or type of equipment. For instance, if it is found that one or more welders are needed at some shop, it is very probable that some can be transferred from another point which has more than can be used to advantage at that particular time, it is only necessary to pull the units to be transferred into a car, block them substantially, and bill to the point where needed. Immediately on receipt they are ready for operation. We have taken advantage of this feature of the plan on a number of occasions and have been able to relieve congestion of work and get engines into service considerably ahead of the time originally estimated. However, officials at all points are very reluctant to surrender any of their welding equipment, claiming that they have not nearly enough as it is.

We have succeeded in making the application of the electric arc welding process on the Rock Island Lines a system proposition not only in the matter of the apparatus itself, but also in its operation. It appeared to us that some roads were using the process in a more or less haphazard manner, leaving the matter to the fancy of the operator. It seemed to us that in going into the practice it would be advisable to throw the whole force of the operating organization into it to get the most out of it. The successful application of the process requires the combination of three factors—engineering knowledge, craftsman's skill, enthusiasm. The direction of the practice of our road rests with the engineering staff of the mechanical department; the actual operation is done by skilled members of the boilermakers', pipefitters', machinists' and blacksmiths' crafts. We do not employ novices or apprentices in this work. Only the best men of the respective crafts are picked for operators. It has been plain from the start that only the highest type of craftsmen could secure the results we want, and we have witnessed the growth of a considerable amount of pride and enthusiasm in the work among our operators. Even with the best of equipment and facilities for welding, we recognize the fact that it is absolutely necessary that only skilled operators be employed.

Under competent direction the skilled and enthusiastic operator will seldom make serious blunders in the application of the process. We have found few of the "know it all" type of men among our skilled craftsmen in this work, whereas I am quite certain some roads have had great difficulties in this line among operators made of green apprentices and "handy men." Further, we have found that the skilled craftsman, who is enthusiastic about the process, is continually finding new and profitable fields for its application.

The connecting link between the engineering staff and the operators is the Supervisor of Welding, who is fully informed on the range of approved

applications and is also the most expert operator on the road. He is continually travelling between the shops keeping the practice of each up to date and seeing that everything runs smoothly. The operators at local points are under the supervision of the foremen and master mechanics in exactly the same manner as lathe operators or other craftsmen.

Due to the shortage of time, it is not within the scope of this paper to discuss in detail the necessary instructions for properly operating a complete welding system. On the Rock Island it was found essential that we compile a complete set of welding instructions, which comprise some 30 typewritten pages. It is the purpose of this set of instructions to standardize the major operations as far as possible. The extreme range of the application of the process has made it quite impossible, up to the present time, to standardize every single operation, but these instructions cover the field in such a general way that the operator is prevented from making serious welding blunders. In view of the fact that our practice has been so rapidly developed by this method, it will be necessary, this fall, for us to revise and reprint our standard welding instructions, which at that time will be very complete. Let it suffice to say that the ability of the operator to make good welds requires that he have some knowledge of metals and their properties, especially as regards expansion and contraction, and that his education by the supervisor be thorough and complete. It is now evident to us that the complete instruction and training of the craftsmen in the art of electric welding should be considered as a necessary additional course to the apprenticeship they served when learning their trades, in order to keep pace with the rapid strides being made today in mechanical engineering practice. It is the desire of the writer and the intention of this paper more fully to acquaint the railway management with the economic value of the new welding process, both in war and in peace, in order that it may be given the great attention that it so well deserves.

The actual results of the operation of the welding equipment and the welding system on the Rock Island Lines have proven very interesting. We have recently undertaken a rather extensive investigation of what the results are, and I am giving overleaf some of the figures we have obtained as a result of about six months' operation of the complete system. The real answer to the question of whether or not the expenditure of some 40,000 dollars for the installation of the system was justified lies in the actual facts—reduction of maintenance cost and actual gain in engine days with our present equipment. In other words, the object of the investigation was to determine whether or not we are saving money by the operation of the system, and whether or not we are actually getting more transportation service out of our motive power and rolling stock equipment. The table overleaf shows some actual figures on the cost of repairing a small number of representative locomotive and other parts (for which we were able to obtain costs at a reasonable expenditure) by the gas and electric method, as compared with the old method, which in many cases involves a complete replacement of the part. It is indeed unfortunate that the railways have not kept detailed costs of the work necessary to the operation of the property. As compared with the old method, the saving of the electric process arises principally in the saving in labor. As compared with the gas process, the electric welding offers a saving in the cost of producing heat and an appreciable saving in cost of labor. It seems that to me under present conditions, and possibly under conditions which will obtain in the future, the labor saving is the most important item. It is evident from the figures that an important saving has been made. In other words, the operation of the process in our shops may be said to have rendered each man more effective, and has enabled each man to accomplish more toward the maintenance of our equipment with the same amount of effort.

Our figures show that the saving effected by the electric arc welding system is being made at the rate of, approximately, 200,000 dollars a year with our present equipment. This figure includes a direct saving as compared with other methods of about 136,000 dollars. The saving arising from the fact that we keep the engines in service a greater proportion of the time makes up the balance of the figure. This saving is being made at the rate of about 1,400 engine days per year.

Another way of looking at the same matter is that by the operation of the electric welding system we have obtained the service of four additional engines, without the additional investment, beyond that required to install the welding system. Four additional engines are worth, approximately, 200,000 dollars. The welding system installed complete cost about 40,000 dollars. The cost of operation of the system for a year is, approximately, 34,000 dollars. Figuring the value of the engines at 40 dollars per day, we will pay for the operation of the whole electric welding system, and will clear 23,000 dollars from this feature of the operation of the electric welding system alone. However, we made important savings, as are shown in the table overleaf, in the repair of parts on engines, where we could not show an actual gain in engine days of service, and this saving amounts to more than twice the saving arising in the increase of the number of engine days of service we get from our equipment. The net return secured on the electric welder investment amounts to, approximately, 500 per cent per annum. The net cost of the installation and equipment per unit under present conditions is, approximately, 1,300 dollars. The foregoing figures show rather conclusively that the installation of the electric welding system has been a profitable investment on the Rock Island Lines.

In spite of the fact that we have probably a larger number of operators than any of the Western roads, we believe that we are far from fully equipped. The field of application of the process is continually widening. We are not able at the present time to handle all the operations which we have demonstrated to be practical and profitable. There is a totally unexplored field of maintenance of freight and passenger cars, which promises to eclipse in importance maintenance of motive power. There is a field in the repair of special track work which we have not gone into up to the present time.

The present indications are very strong that when we go fully into the electric welding process in fire box boiler, locomotive, machinery, steel tanks, car work, track work, &c., that we can well use 150 units and effect a net saving, as I have stated before, of approximately 1,000,000 dollars a year. Within the last three years the arc welding process has been greatly improved and developed, both in the equipment for making the weld and in the welding material. It has, in fact, been developed to such a state that it will, no doubt, cause changes in many forms of construction, and the welding of fire boxes, tanks, &c., will become an economical practice. In fact, it is in the car field of construction and maintenance that we may look for a wonderful development. For instance, today the cast-steel truck side frame will last almost indefinitely where the electric welder is used in maintenance. Also it is as a preventive that the electric welder has a great field by the intelligent application of spot welding with the electric arc. It will, no doubt, be possible to tie down bolts and nuts in the various parts of the rolling stock and motive power in such manner as to prevent their working loose, with the attendant very large saving in maintenance and operating expense. Only recently we have been able by using what is called a slag coated electrode, to deposit steel having a carbon content of 0.50 per cent, which will enable us to do some work which we have been unable to do heretofore. We can successfully take care of the worn or damaged flanges of driving wheels, and should be able to reclaim much of the special work and rail steel by successfully building up the worn parts or broken sections.

TABLE I.—Comparison of Electric Welding versus Old Methods and Gas Welding

Description of Parts.	Cost Old Method.	Cost Gas Welding.	Cost Electric Welding.	Saving Over Old Method.	Saving Over Gas.	No. Engines.
	£	£	£	£	£	
Valve stems	16·28	15·36	4·76	11·52	10·60	6
Eccentric straps	17·84	7·92	3·96	13·87	6·95	2
Cylinder cocks	1·30	·104	·34	1·02	·70	1
Cross heads	535·68	120·72	37·72	513·07	82·99	13
Piston heads	47·52	25·74	10·34	37·20	15·40	4
Motion studdies	8·91	3·96	1·44	7·47	2·52	2
Frame braces	32·50	16·60	10·03	24·50	13·99	10
Crank arms	18·81	20·16	3·15	15·67	17·01	6
Rock box castings	4·59	7·29	2·43	2·16	4·86	1
Transmission bar	2·52	4·32	1·73	1·13	3·09	2
Reach rod	1·80	1·80	·54	1·26	1·26	1
Rocker arms	29·75	13·24	4·24	16·51	9·00	4
Engine truck equalisers	7·70	17·24	4·32	3·38	12·92	2
Truck frame	15·00	12·00	4·64	11·36	9·60	3
Trailer jaws	1·70	4·32	1·44	·26	2·88	1
Extension piston cross head	6·30	4·32	1·44	4·86	2·88	1
Brake beams	1·80	2·16	·48	1·32	1·68	1
Brake hangers	1·19	7·46	1·70	·49	5·76	1
Smoke arc brace	3·60	4·32	2·16	1·44	2·16	1
Air pump valves	2·50	1·33	·55	1·95	·78	1
Lags on valve yoke	62·65	21·60	3·60	59·05	18·00	6
Push bar wheels	10·00	10·00	2·00	8·00	8·00	4
Stillson wrench	1·20	1·00	·24	·96	·76	1
Drill chuck	15·00	2·15	·65	14·35	1·50	1
Driver brake fulcrum	3·12	·72	·72	2·40	·00	1
Wheel spokes	1,570·00	113·00	28·00	1,542·00	85·00	15
Main-rod blocks	15·33	23·31	6·42	7·04	16·89	9
Triple valve gauge	29·00	5·27	1·58	27·42	3·69	1
Link blocks	79·04	51·60	15·48	63·56	36·12	20
Lift shafts	23·98	4·32	1·08	22·90	3·24	1
Quadrant	7·40	11·00	3·00	4·40	8·00	3
Wedges	55·05	60·00	17·00	38·05	43·00	25
Chafing castings	5·00	10·70	2·00	3·00	8·70	1
Plugging and bolting up holes	340·00	230·00	140·00	200·00	140·00	70
Tire rim keys	2·22	·72	·72	1·50	·00	1
Throttle stem	1·50	1·50	·36	1·14	1·14	1
Reverse lever support	3·00	2·50	1·00	2·00	1·50	1
Smoke box	61·20	32·63	9·00	52·20	23·63	2
Hub liners	12·51	12·11	3·11	9·40	9·00	2
Strip on cross heads	25·00	12·00	12·00	13·00	00·00	1
Fire door handle	1·75	1·09	·34	1·41	·75	1
Boiler casings	65·21	30·80	9·52	55·69	20·80	1
Frame buckle	3·00	2·11	·81	2·19	1·30	1
Trailer yokes	2·25	1·95	1·95	·30	·00	1
Motion bridle	9·33	10·17	2·17	7·16	8·00	1
Combination lever	1·92	·56	·16	1·76	·40	1
Lags on trailer axle	4·37	1·20	·30	4·07	·90	1
Centre castings	76·21	30·00	9·00	67·21	21·00	3
Spring blocks	1·15	1·00	·34	·81	·66	1
Guide blocks	5·65	2·97	1·39	4·26	1·58	1
Binder	15·10	15·10	3·00	12·10	12·10	2
Steam pipes	3·78	1·08	·36	3·42	·72	1
Flat spots on tyres	90·00	90·00	30·00	60·00	60·00	4
Cylinder bushings	5·00	5·00	1·00	4·00	4·00	1
Building up side rods	21·60	21·60	3·60	18·00	18·00	2
Grease cups	11·79	11·43	3·96	7·83	7·47	5
Stationary fire door	9·00	9·00	3·00	6·00	6·00	1
Cracks in tanks	272·00	113·00	33·16	238·84	79·84	14
Petticoat pipes	140·00	55·00	15·00	125·00	40·00	10
Filling worn spots	2,657·00	1,664·00	328·00	2,329·00	1,336·00	123
Pins	70·00	27·00	7·00	63·00	20·00	27
Reverse lever parts	388·00	74·00	30·00	73·00	51·00	36
Total	96,694·19	32,756·76	9921·61	86,618·58	51,834·19	

TABLE II.—*Comparison of Electric Welding, Without Other Methods.*

Parts that cannot be Welded by Gas.

Description of Parts.	Cost of Other Methods.	Cost of Electric Weld.	Saving.	No. Engines.
	£	£	£	
Pedestals	845·00	45·84	800·76	5
Tank frames	9·03	1·38	7·67	1
Shop boilers	54·50	13·40	40·50	1
Piston rods	79·94	19·27	62·27	30
Sharp flange driverts	185·40	50·24	145·16	
Truck side	194·00	10·30	183·80	4
Building up cr. axles	121·80	4·90	116·90	1
Steel car underframe	11·36	1·71	9·65	1
Building up car axles	818·00	25·24	792·76	
Cracks in steam boiler	226·68	72·74	231·12	24
Welding Steel	2,407·65	822·80	2,585·15	122
Frames	961·45	133·20	797·75	11
Cracks in the axle boxes	2,451·27	297·17	2,154·10	82
Total	67,830·15	61,154·42	96,694·73	

TABLE III.—*Summary.*

Costs and Savings—Per Month.

Cost of Other Methods.	Cost of Gas Weld.	Cost of Electric Weld.	Saving Over Other Methods.	Saving Over Gas Weld.
£	£	£	£	£
6,434·10	2,756·76	921·61	5,512·49	1,834·13
7,839·15	3,987·62*	1,154·15	6,684·75	2,843·00*
14,273·25	6,488·10	2,075·06	12,197·24	4,377·13

Costs and Savings—Per Year.

Cost of Other Methods.	Cost of Gas Weld.	Cost of Electric Weld.	Saving Over Other Methods.	Saving Over Gas Weld.
£	£	£	£	£
77,205·00	33,060·84	11,053·92	66,140·92	22,009·56
94,060·80	47,850·04*	13,853·04	80,216·76	30,516·00*
171,272·00	77,437·88	24,911·30	146,356·84	52,525·56

* Figures show cost of gas weld if work could have been welded with gas.

It has been our purpose in establishing the practice in the welding field to look the facts squarely in the face and apply either the gas or electric process, depending on which shows the best results at the lowest price. At the present time we are of the opinion that the electric process will supersede the gas process on all steel welding and some of the rough-steel cutting. In the cutting of boiler steel and all close cutting, however, and the welding of cast-iron and the non-ferrous metals, the gas process has unequalled advantage. We are operating 75 gas torches and one acetylene generating plant on the same general principle as obtained in the case of the electric arc welding equipment. It is also best to use gas welders at all points where only occasional welding is done and which would not justify the investment necessary for the installation of electric welder.

While we have not been able to meet the suggestion of the American Railway Association's Special Committee on National Defence, which reads as follows: "About 15 per cent of locomotives are ordinarily under repair. If this percentage were reduced to 10 per cent, which figure has been reached by some roads, it would mean an addition of 3,325 locomotives to the number in service," we have been able to move in that direction and have nearly reached the figure of 10 per cent, and at the same time added to the profits of the road by our welding system. It is our belief that with approximately five times the amount of electric welding capacity we have at present, we can show at least five times the annual net saving, which would amount to 1,000,000 dollars a year, and that we can, with this equipment in operation, show a saving of around 7,000 engine days per year, which means that we would be able to secure from our present engines a mileage that will equal that which could otherwise only be secured by the purchase of 23 additional engines.

With this figure in mind, and knowing that the Rock Island Lines have, approximately, 1,600 locomotives in service, it is easy to realize the enormous possibilities in the direction of increasing the service from motive power equipment on the combined railway systems of the nation, both as an economic proposition and as an aid in war, were they all equipped with welders. The demand for transportation service is so urgent, and the effectiveness of the nation in this war depends so directly upon the quantity of service which may be rendered, that it seems imperative that the railroads should go into the welding field on a large scale at the earliest possible moment. These figures have been given with particular reference to motive power units. While we have no tangible data on what may be accomplished in the rolling stock field, it is our belief, from the preliminary survey of the situation, that even a greater gain can be made toward keeping the equipment in service in this field than has been made on the motive power units.

The application of the new welding processes is important now as never before, when every railroad must get the maximum of service from all of its equipment; when the nation demands the supreme effort of the transportation system of the country.

What I have just said refers particularly to railroad work and gives some idea of what we think can be accomplished, using as a basis for estimation the results we have actually accomplished on the Rock Island Lines. The enormous possibilities, however, in the application of the new welding processes to other fields are of absorbing interest in view of the present world conditions as a result of war and their effect on labor and material. Not least among the possibilities of the welding processes is that of substituting welding for riveting in bridge and building steel construction. These structures are permanent, and knowing the safety of the welded joint, from our experience with boiler works, &c., we can safely prophesy that such structures will be welded in the future in place of being riveted, from the standpoint of both economy and safety. This

is a new idea, but when we consider that five years ago the complete welding of a locomotive fire box was considered an absurdity, the newness of the idea of welding steel structures is not in the least appalling.

The possibilities in the field of ship construction and repair have been explored to some extent. Every harbor on the Atlantic coast and the large harbors on the Pacific coast are all equipped with the apparatus. Up to the present time the welding, however, has been entirely on repair work, but the savings effected by the use of the process on boiler repairs, repair of broken stern posts, rudder shoes, propeller-shaft sleeves and miscellaneous superstructure work, have been important, and enormous economies have been made. It is sometimes possible to make necessary ship repairs in one day with the electric welder that otherwise would have required two weeks, and since the value of a ship at sea per day ranges anywhere from $500 to $2,000, it can readily be appreciated that there is an enormous demand for the work at the present time. The commercial welders in the Atlantic ports charge from $10 to $20 per hour for the services of one man, which is a very good indication of the fact that the application of the process is valuable to the steamship owners. It is the understanding that at the present time welding in place of caulking on torpedo boat destroyer hulls enables the hull to offer much greater resistance to shock of collision, shell fire, or torpedo destruction than it otherwise would. It is reasonable to assume this practice will be extended in our navy and merchant marine in the welding of ship's hulls. Already several of the shipbuilders on the Atlantic coast have made application for patents on the completely welded hull, and while no actual construction of this nature has been done as yet, it is probable in the near future that we shall see the completely welded hull in service.

The extensive application of the arc welding process in the United States has led other countries to the adoption of the improved practice. Notable examples of this are the introduction of welding outfits into the principal harbors of the South Atlantic. In the German and English naval engagement off the Falkland Islands, the British ships were severely damaged and put into a port for repairs, where they obtained a marine electric welding outfit, immediately putting to sea and effecting their repairs in a comparatively short time; thus their ships were ready again for service in a much shorter time than would otherwise have been possible, when every minute was very valuable indeed.

The electric welding equipments were extensively used in repair work during the construction of the Panama Canal. The South American Railway Corporation and some mining corporations are installing the portable equipment manufactured in this country. A Japanese Government expert has recently returned to Japan with complete information and the introduction of the process will probably be rapid in that enterprising country. Already a considerable amount of equipment has been shipped to Spain, Norway and Sweden for use on railways and in steel foundries.

Four portable electric arc welding units such as are in use on the Rock Island lines are now being shipped to the French fighting front for use on military railways by the "American engineer regiments (railways)." The application here covers not only the standard guage railways which transport ammunition and supplies to the front lines, but also in the construction and maintenance of narrow-gage feeder and trench railways, the tracks of which are ordinarily constructed of small steel rails which may be welded to the steel ties, these railways being of a portable type. There are also a very large number of automobiles, motor cycles, etc., to be repaired, as they are constantly breaking down, due to the extremely strenuous service in which they are employed. In fact, without the use of the gas and electric welders, the life of such equipment would be very short indeed, when used at the battle front. Within the last year we have seen the increasing use of

the English tanks, which call for very extensive repairs, that can be most efficiently made with the electric welder. The artillery also makes a heavy demand on the welders for repairing gun carriages, gun mechanism and the guns themselves. It is necessary on the front that these repairs be made expeditiously under most trying conditions—conditions that it would be practically impossible to meet were it not for the autogenous welding devices.

A little to the rear of the actual fighting front, the electric welders are in demand for repairing horse equipment, including the repairing of vehicles, and it is thought that in the near future it can be used for welding new calks on the horseshoes to replace those that are worn, without the necessity of removing the shoe from the horse's hoof. The strenuous military campaigns require frequent demolition of iron and steel structures, including heavy guns, and the electric and gas cutting processes have been found extremely valuable owing to the great speed at which iron or steel structures may be demolished beyond the hope of repair.

In many instances there is a source of electric power supply from the power lines that run in or through the theater of war. However, in some instances where no such lines are encountered and for some reason or other it is not possible hastily to construct a power line, it becomes necessary to set up a small internal-combustion, power-driven unit in order to secure an electric power supply, which is becoming a necessity in modernized warfare. Electric welders can be used for trench lighting and searchlight work as well as for welding, which makes them very valuable equipment. One of the particular advantages in using the electric welder is that it eliminates the necessity for transferring welding gas, which would many times be practically impossible.

The final result of every effort must be to put effective fighting men in the field because it is increasingly evident that the war will be won by the men who go over the top and put the enemy out of business, by exterminating him. It is easy to lose sight of the fundamental facts in this connection by a lot of superficial talk about the economic side of the war. We are so far removed from the active fronts that it is easy to lose ourselves in a lot of talk about industrial economics and about how food will win the war, how locomotives will win the war, how sugar will win the war, how torpedo boat destroyers will win the war. It is easy to forget that it is actually the fighting man-power we can bring to bear upon the enemy that will really be the deciding factor. We must have the greatest possible industrial efficiency that can be obtained regardless of cost, regardless of the effort required to obtain it, and the reason is that we have fighting men on the front who must have everything that is required to keep up their morale and keep them in the very highest degree of effectiveness. Industrial efficiency is a part of our job, a necessary part of the nation's job of backing up the armed forces in the field. The big job is done by the men who go over the top or over the seas and exterminate the enemy.

Another interesting application of the electric arc process is in the repair of the ships. Every ship that is seaworthy is in service at the present time, and time for repairs must be cut to the irreducible minimum. The electric arc process is used extensively for boiler repairs and repairs about the deck and superstructure. It is not unusual to make repairs to a ship's boiler with the electric arc process in 24 hours while a ship is unloading her cargo which otherwise would require a week to ten days to make and fifty times the cost. The use of the oxy-acetylene and the electric arc processes on the ex-German ships which were badly smashed up by the Germans allowed these ships to be put into the service of the United States from four to six months sooner than would have been possible otherwise.

In the steel foundry the electric arc process is used to repair defects in steel castings which would otherwise make the casting useless. At the

present time, the demand for steel castings for the Government is greater than all of the steel foundries of the country can supply. The labor conditions in this industry are generally in such conditons that it is impossible to get sufficient highly-skilled labor to make perfect castings. The net result of the use of the welding process is to get the same output of castings as if all the men were skilled craftsmen.

The electric arc process is being used in boiler shops to replace riveting, a seam may be welded cheaper than it can be riveted, and the joint is stronger than it is possible to obtain with the riveted construction in making a joint. One man with an electric arc will do as much work as a riveting gang of four men and the electric power required to operate the electric welder is about the same as the power required to run a pneumatic hammer.

Another interesting application in the process is in oil refineries, where it is used both in addition to riveting and in place of riveting. On some of the large stills whose operation must be as nearly continuous as possible, the riveted joints are welded instead of being caulked. Such a joint is stronger than the original plate and the life of such a still is more than doubled.—"Engineering."

APPLICATION OF ELECTRIC WELDING TO SHIPBUILDING.

When the history of the development of engineering during the war comes to be written, the chapters on the progressive improvement of electric welding will afford much pleasure to those who believe that the inventive spirit is yet very much alive in this country.

Welding has been applied in countless ways to the manufacture of munitions of war, and the more advanced shipbuilders soon recognized that this method of construction offered almost endless possibilities to shipbuilding. When the matter was brought under the notice of Lloyd's Register, the committee, with a characteristic desire to afford the shipbuilding community the very latest technical information on the subject, arranged to carry out an exhaustive series of tests, which have extended over a period of six months.

Although electric welding in various forms has been employed for many years for ship repair work, yet, in practice, owing to many factors its use has been practically confined to those parts of the structure which are not likely to be exposed to important structural stresses. It is only in recent times, commencing from the early days of the war, that appreciable progress has been made in the developments of electric welding which would appear to justify the extension of such methods to replace the usual riveted connections of heavy structural work.

The aim has been to secure reliability and regularity of operation in the welding process, and to assist the workmen by improving the means of control over the work. Just as in the case of application of steel to shipbuilding, it was necessary to devise means for the production of the material in large quantities and of constant quality, so also is it necessary that the welding electrodes should be manufactured with the greatest possible degree of uniformity. Reliability of operation is also facilitated by adjusting the density of the electric current to the size of electrode used, and further, the size of the electrode should reasonably vary directly with the thickness of material to be connected.

Research was necessary to discover means for minimizing the burning of the deposited material in the direction of preventing oxidation. In the early days of welding, the molten electrode was exposed to air throughout the whole time of deposition and, consequently, oxidation was more or less

certain to occur. With coated metal electrodes, burning is reduced to a minimum by the use of a slag which envelops the molten steel and floats on its surface after contact is obtained with the material to be connected. Even in this system skilled workmanship is essential, as the production of a long arc obviously increases the chances of burning.

The composition of the material of the electrode in relation to the nature of the steel to be connected is obviously a matter of importance. What the composition is to be can only be gaged by experiment and by wide experience, and it is in devising the physical tests for work of this nature that the greatest difficulties arise.

It is commonly accepted that the tests imposed on manufactured material do not in any way represent the strains which may be experienced in practice. Such tests are rather based on simple means for determining the average reliability of the material. Thus also in this case no one particular test is likely to determine whether the welding process under trial is sufficient for the work it is likely to have to do. It is, therefore, necessary to approach the problem rather on the basis of circumstantial evidence, and to decide from a number of different types of experiments whether, on the whole, the performance is satisfactory.

The more particular problem in shipbuilding is the connection of mild steel containing a percentage of carbon of about .15. This material in the form of plates and section bars has considerable work done to it during the process of manufacture, with the consequence that it possesses a fine structure and a ductility which is uniform in every direction. The finished material may be said to be practically free from fibrous structure.

With electric welding, molten metal is attached to the mild steel and from the extent of the cooling surface the deposited material is rapidly lowered in temperature with the consequence that the weld tends to become deficient in ductility. The problem therefore is to select the material of the electrode so that the general elastic properties of the structure are not unduly depreciated.

The investigations were undertaken to determine the possibilities of the application of electric welding to shipbuilding, and as it was desired to obtain as good a knowledge as possible of the physical properties of the combination of rolled and welded material highly skilled operators were employed.

It must, therefore, be realized that the results of the experiments which have been made represent skilled practice, and that in general such performance can only be equalled with good workmanship and efficient supervision.

NATURE AND DESCRIPTION OF EXPERIMENTS.

The general scope of the experiments included:

(*a*) Determination of modulus of elasticity and approximate elastic limit.

(*b*) Determination of ultimate strength and ultimate elongation.

(*c*) Application of alternating stresses with—

(1) rotating specimens,

(2) stationary test pieces.

(*d*) Minor tests, such as—

(1) cold bending of welds,

(2) impact tests of welded specimens.

(*e*) Chemical and microscopic analysis.

Tests were carried out on specimens as large as possible, particularly in respect of the static determinations of elasticity, ultimate strength and elongation, some of the test specimens being designed for a total load of just under 300 tons. The advantage of these large specimens was that the

effect of workmanship was better averaged and the results were more comparable to the actual work likely to be met with in ship construction.

With alternating stresses the specimens were relatively of small size. For the rotating test pieces, circular rods, mainly machined from a welded plate, were used, the diameters selected being 1 inch and ¾ inch. These bars, about 3 feet in length, were attached to a lathe headstock and a pure bending moment in one place was applied by means of two ball races to which known weights were attached. The material of the bar was thus exposed alternately to maximum tension and to equal maximum compression once in each revolution. The machine was run at about 1,060 r.p.m. Bars of identical material were tried in pairs, one specimen welded and the other unwelded, and the number of revolutions before the specimens parted was observed for various ranges of stresses varying from ± 15 tons to ± 6 tons.

In the second series of alternating stress experiments flat plates were used of three thicknesses, viz.:—¼-inch, ⅜-inch and ½-inch. These specimens were tried in groups of 4, each group consisting of 1 plain, 1 butt welded, 1 lap welded, and 1 lap riveted plate. The specimens, which were about 14 inches long by 5 inches broad, were clamped along the short edges, so that the distance between the fixed lines was 12 inches. Each plate was also clamped, near the middle, to the end of a pillar, which by means of a crank arm was caused to oscillate and to bend the specimen equally up and down by adjustable amounts (the maximum total movement in any of the experiments tried was 5/16 inches). The machine was run at various revolutions (not exceeding 90 per minute), and the number of repetitions at which the specimen parted was observed.

Minor tests of various kinds were undertaken, of which the principal had reference to the suitability of the welded material to withstand such bending and shock stresses as might occur in the shipbuilding yards. The experiments on bending consisted of doubling the welded plate over a circular block of diameter equal to three times the plate thickness, and comparing the results with those of the plate of the same material, but unwelded.

In the impact tests heavy weights were dropped from various heights on to the welded portion of a plate 5 feet in length and 2 feet 6 inches in breadth, the weld being across the plate parallel to the shorter edge. The deflections were noted and the condition of the weld was examined after each blow.

The chemical and micrographical examination followed the ordinary practice.

SUMMARY OF EXPERIMENTAL RESULTS.

1. *Modulus of elasticity and approximate elastic limit.*

(*a*) In a welded plate the extensions in the region of the weld are sensibly the same as for more distant portions of the unwelded plate.

(*b*) With small welded specimens containing an appreciable proportion of welded material in the cross-sectional area, the relation between extension and stress is practically the same, up to the elastic limit, as for similar unwelded material.

(*c*) The elastic limit (or the limiting stress beyond which extension is not approximately directly proportional to stress) appears to be slightly higher in welded than in unwelded material.

(*d*) The modulus of elasticity of a small test piece, entirely composed of material of the weld, was about 11,700 tons per square inch, as compared with about 13,500 tons for mild steel and about 12,500 tons for wrought iron.

2. *Ultimate strength and ultimate elongation.*

(*a*) The ultimate strength of welded material with small specimens was over 100 per cent of the strength of the unwelded steel plate for thicknesses of ⅛ inch, and averaged 90 per cent for plates of ¾ inch and 1 inch in thickness.

(*b*) Up to the point of fracture the extensions of the welded specimens are not sensibly different from those of similar unwelded material.

(*c*) At stresses greater than the elastic limit, the welded material is less ductile than mild steel, and the ultimate elongation of a welded specimen when measured on a length of 8 inches, only averages about 10 per cent as compared with 25 per cent, to 30 per cent for mild steel.

3. *Alternating stresses.*

(*a*) Rotating specimen (round bar):

(1) Unwelded turned bars will withstand a very large number of repetitions of stress (exceeding, say, 5,000,000), when the range of stress is not greater than from 10½ tons per square inch tension to 10½ tons per square inch compression.

(2) Welded bars similarly tested will fail at about the same number of reversals, when the range of stress exceeds ± 6½ tons per square inch.

(*b*) Stationary test pieces (flat plate):

(1) Butt-welded specimens will withstand about 70 per cent of the number of repetitions which can be borne by an unwelded plate.

(2) Lap-welded plates can endure over 60 per cent of the number of alternations necessary to fracture a lap-riveted specimen.

4. *Minor tests.*

(*a*) Welded specimens are not capable of being bent (without fracture) over the prescribed radius to more than about 80 degress with ¼-inch plate, reducing to some 20 degrees where the thickness is 1 inch. Unwelded material under the same conditions can be bent through 180 degrees.

(*b*) Welded plates can withstand impact with a considerable degree of success; a ½-inch plate of dimensions already quoted sustained two successive blows of 4 cwt. dropped through 12 feet, giving a deflection of 12 inches on a length of about 4 feet 6 inches without any signs of fracture in the weld.

5. *Chemical and microscopic analysis.*

(*a*) Chemical analysis.

(1) The electrode was practically identical with mildsteel, but there was a greater percentage of silicon.

(2) The material of the weld after deposition was ascertained to be practically pure iron, the various other contents being carbon, .03; silicon, .02; phosphorus, .02; and manganese, .04 per cent, respectively.

(*b*) Microscopic examination.

(1) The material of the weld is practically pure iron.

(2) The local effect of heat does not appear largely to affect the surrounding material, the structure not being much disturbed at about 1/16 of an inch from the edge of the weld. The amount of disturbance is still less in thin plates.

(3) The weld bears little evidence, if any, of the occurrence of oxidation.

(4) With welds made as for these experiments, *i. e.*, with flat horizontal welding, a sound junction is obtained between the plate and the welding material.

6. *Strength of welds (large specimens).*

(*a*) Butt welds have a tensile strength varying from 90 per cent to 95 per cent of the tensile strength of the unwelded plate.

(*b*) Lap welds.

(1) With full fillets on both edges the ultimate strength in tension varies from 70 per cent to 80 per cent of that of the unwelded material.

(2) With a full fillet on one edge and a single run inferior to those where a full fillet is provided for both edges.

(*c*) Riveted lap joints: For plates of about ½ inch in thickness, the specimens averaged about 65 per cent to 70 per cent of the strength of the unperforated plate.

OBSERVATIONS ON EXPERIMENTAL RESULTS.

(1) *Static elasticity.*—It will be observed that the statical tests made to determine the elasticity indicate that, in general, the combination of welded and unwelded material behaves practically homogeneously up to at least the elastic limit. Moreover, the experiments show that the process of welding is such that the stress is distributed practically uniformly over the weld, and also transmitted uniformly to the adjacent plates.

The material of the weld is practically pure iron, and from the tests made on a specimen composed entirely of the deposited material of a weld, it will be seen that for given stress the weld stretches slightly more than mild steel. This property will enable any undue occurrence of load to be transferred in a proper manner to adjacent portions of the structure. When, however, the stress exceeds the elastic limit and is so great that the extension grows continuously without increase of load, the welded material fails sooner than mild steel. This disadvantage is, however, of little practical importance in shipbuilding, and may be regarded as negligible in the particular problem under consideration.

(2) *Dynamic elasticity.*—In a structure, such as a ship, which is exposed to variations and reversal of stresses, it is extremely important to know whether the material to be used is likely to break down rapidly under such alterations and ranges of stress as are likely to be experienced. The modified Wohler tests employed in the experiments certainly indicate, if considered solely by themselves, that whereas for a given number of alternations mild steel would withstand a range of stress of, say, ± 10½ tons, the welded material might be expected to fail at about ± 6½ tons, a figure which is more nearly experienced in ordinary ship construction.

It would appear to be necessary to design the welded joints in such a manner that the amount of work likely to be thrown on the joint is as small as possible, and to meet such a condition a welded joint requires to be either lapped or strapped.

It will be noticed that the material in the weld appears to be nearly pure iron, and experiments of repetitive stress show that wrought iron bars are likely to fail under a range of stress of perhaps ± 7 tons to 8 tons, as compared with mild steel at ± 10 tons to 11 tons. The weld has to be deposited electrically and is subject to variations in workmanship; it would consequently be considered satisfactory if the material could withstand a range of stress of, say, ± 6½ tons.

Consideration of the dynamic elasticity of properties appears to show that in any case the welded material can experience as large a number of repetitions of stress as wrought iron could do, and it is always recognized that although iron could not approach the tests for mild steel, yet it was a satisfactory material for shipbuilding purposes. Further, attention to design of detail will increase the performance of the welded joint, and in addition it must not be forgotten that 5,000,000 repetitions of stress is perhaps more than equivalent to 10 years' good sea service.

(3) *Physical Nature and Properties.*—It has been mentioned that the welds experimented with are to be regarded as having been produced

under most favorable conditions, and that throughout the experiments welds were made with the specimens horizontal and below the operator. In practice, welds will require to be made vertically and overhead as well, consequently, extreme care will be required in such operations.

The physical examinations indicate that the materials of the electrode and the system of welding adopted were suitable and reliable. Moreover, there was little apparent oxidation and the material in the neighborhood of the weld was not affected to any prejudicial extent.

(4) *Strength of large welds.*—Broadly speaking, the tensile strength of butt welds was as great as the unwelded material, but it is considered that greater reliability of workmanship is obtained with joints which are either lapped or strapped.

It was also found that the lapped joint was practically as strong as a riveted lapped joint and would probably remain tight when subjected to more trying conditions than are necessary to disturb a riveted lap joint.

A meeting of the technical committee of Lloyd's Register to consider the application of the results of these tests, was held on the 25th ult., when it was decided that there appeared to be sufficient justification for the extension of the process of arc-welding to the connection of the main structural portions of ships. Such application should, however, only proceed cautiously in view of the unknown factors involved, in the development of the most important of which are the need of experience with the details of the welded joints, and the necessity of training skilled workmen and supervisors.

After full consideration of the subject the technical committee decided to recommend to the general committee that provisional rules, as proposed by the chief ship surveyor, should be adopted for the application of electric arc-welding to shipbuilding, and that the approval of the society should be given to any process which complies with such regulations.

A special meeting of the general committee was held on Thursday, the 15th inst., to receive the report of the technical committee, and it was decided that the subjoined regulations should be adopted as a tentative measure.

TENTATIVE REGULATIONS FOR THE APPLICATION OF ELECTRIC ARC WELDING TO SHIP CONSTRUCTION.

(a) System of Welding and Workmanship.

1. The system of welding proposed to be used must be approved and must comply with the regulations and tests laid down by the committee.

2. The process of manufacture of the electrodes must be such as to ensure reliability and uniformity in the finished article.

3. Specimens of the finished electrodes, together with specifications of the nature of the electrodes, must be supplied to the committee for purposes of record.

4. The committee's officers shall have access to the works where the electrodes are manufactured, and will investigate, from time to time as may be necessary, the process of manufacture to ensure that the electrodes are identical with the approved specimens.

5. Alterations from the process approved for the manufacture of electrodes shall not be made without the consent of the committee.

6. The regulations for the voltage and amperage to be used with each size of electrode, and for the size of electrode to be employed with different thicknesses of material to be joined, are to be approved by the committee.

7. The committee must be satisfied that the operators engaged are specially trained, and are experienced and efficient in the use of the welding system proposed to be employed.

8. Efficient supervisors of proved ability must be provided, and the proportion of supervisors to welders must be submitted for approval.

(*b*) *Details of Construction.*

9. The details of construction of the vessel and of the welds are to be submitted for approval.

10. Before welding, the surfaces to be joined must be fitted close to each other and the methods to be adopted for this purpose are to be approved.

11. All butt and edge connections are to be lapped or strapped.

12. With lapped connections the breadths of overlaps of butts and seams and the profiles of the welds are to be in accordance with the following table:

Thickness of plate. Inches.	Width of overlap seam and butt. Inches.	Total thickness. Inches.
.40 and under	2¼	.28
.60	2½	.38
.80	2¾	.48
1.00	3	.50

Intermediate values may be obtained by direct interpolation, and for thicknesses below .40 the throat thickness is to be about 70 per cent of the thickness of the plate.

13. A "full weld" extends from the edge of a plate for a distance equal to the thickness of plate to be attached, and the minimum measurement from the inner edge of plate to the surface of weld is the throat thickness given in the table above.

14. A "light closing weld" is a single run of light welding worked continuously along the edge of the plate. Such a weld may, however, be interrupted where it crosses the connection of another member of the structure.

15. An "intermittent or tack weld" has short lengths of weld which are spaced three times the length of the weld from center to center of each short length of weld. Such tack welding may vary in amount of weld between a "full weld" and a "light closing weld."

16. The general character of welds is to be in accordance with the following table:

	Inside edge.	Outside edge.
(*a*) Butts of shell, deck and inner bottom plating	F	F
(*b*) Butts of longitudinal girders and hatch coamings		
(*c*) Edges of shell, deck and inner bottom plating	L	F
(*d*) Butts and edges of bulkhead plating		
	Toe.	**Heel.**
(*e*) Frames to shell, reverse frames to frames and floors	T	L
(*f*) Beams to decks		
(*g*) Longitudinal continuous angles		
(*h*) Side girders, bars to shell, intercostal plates, floors and inner bottom		
(*i*) Bulkhead stiffeners		

F = full weld, L = light weld, T = tack weld.

17. All bars required to be watertight are to have continuous welding on both flanges with tack welding at heel of bar.

18. The welded connections of beam, frame and other brackets are to be submitted for special consideration.

19. The committee may require, when considered necessary, additional attachment beyond that specified above, and the welding of all other parts is to be to their approval.

As is customary when a new process of construction, of which experience in conditions analogous to those which are met with in ships is not available, is proposed to be adopted in vessels classed by Lloyd's Register, the committee have decided that the classification will be subject to the notation "experimental," and a record of "electrically welded" will also be made in the society's register book.

It is unnecessary to enlarge on the importance of the step now taken, and the rivetless ship, at one time considered only as an unattainable ideal, is now well on the way to becoming an accomplished fact.

The prompt and energetic manner in which the possibilities of electric welding have been investigated by Lloyd's Register is worthy of the best traditions of the society, and the vigor and initiative of the present regime bids fair to secure for that Institution an even more preponderating position than it has filled in the past—"Engineering."

AUTOGENOUS WELDING AS ALLOWED BY BOARD OF SUPERVISING INSPECTORS.

The following regulations relative to welding and reinforcing boilers have been announced by the Board of Supervising Inspectors of Steam Vessels of the United States Steamboat Inspection Service:

All calking edges on internally-fired boilers may be reinforced by these processes.

Calking edges of the shells of externally-fired boilers, above the fire line only, may be reinforced.

Cracks extending from edge of lap to rivet, except on seams below the fire line in externally-fired boilers, may be welded.

Cracks not exceeding 30 inches in length in back connection sheets, wrapper sheets, bottoms of combustion chambers, heads and other stayed surfaces may be repaired by welding.

Where cracks are repaired by welding, holes shall be drilled entirely through the plate at each extreme end of the crack, except in small cracks from rivet to calking edge.

Circumferential or lengthwise cracks not exceeding 16 inches in length in plain or corrugated furnaces may be welded.

Where plates in back sheets of back connections, wrapper sheets of sides and bottoms of back connections of any boilers, side sheets and legs of furnaces and bottoms of furnaces of firebox boilers, and other stayed surfaces are reduced in thickness not exceeding 40 per cent of the original thickness, they may be reinforced, such reinforcing not to exceed an area of 200 square inches in any one plate.

When such reinforcing extends over stays and braces, such stays and braces shall come completely through the reinforcing so as to be plainly visible to the inspectors.

When the corroded portion of stayed or riveted surfaces of the back sheets or wrapper sheets or bottoms of back connections of any boilers, or side sheets and bottom sheets of furnaces or legs of firebox boiler exceeds 300 square inches, the same may be repaired by the removal of the corroded portion and the replacement thereof by a new piece of plate the edges of the new plate being welded in position.

Stay-bolts, braces or rivets shall pass through the body of the new plate as before, the area of the new piece not to exceed 24 inches by 24 inches, or 30 inches, in any one direction, the welded edges to be V'd or beveled along the joint prior to welding.

THIN PLATES MAY BE REINFORCED.

Where plates of shells and other parts of internally-fired boilers subject to tensile strain are reduced in thickness by corrosion not to exceed 25 per cent of the original thickness, they may be reinforced, such reinforcing not to exceed an area of 200 square inches.

Where calking edges and laps have been reinforced, local inspectors shall require the rivets to be cut out and redriven if they find by inspection that it is necessary.

No welding shall be allowed in cracks in the shell plates or other plates subject to tensile strain.

Cracks extending through rivet holes in single-riveted or double-riveted seams in stayed surfaces of back connections of any boilers or side sheets, of legs or bottoms of firebox boilers which are stayed surfaces, may be welded up to a length of 6 feet exclusive of rivet holes.

Where cracks extend through rivet holes in stayed surfaces, the piece extending from the rivet to the edge of the lap may be removed where convenient to do so, and the place where the piece has been removed may be replaced by being built up and reinforced by either of these processes.

Where leaks develop around stay-bolts and the stay-bolts are otherwise intact, the nuts may be removed from the ends of the stay-bolts and the stay-bolts may be welded into the shell by welding a beveled collar or ring around the stay-bolt. The width and depth of such collar shall equal one-half of the diameter of the stay-bolt. In all such cases of applying welding rings or collars around stay-bolts, the material shall be hammered while in a glowing state as it is applied.

In all cases where metal is deposited on stayed surfaces, the operator shall hammer, when practicable, the deposited metal while it is in a glowing state.

No repair work by any welding process shall be allowed until coupons showing the character of the work proposed to be done by the applicant have been tested and submitted, together with an explanation and report of the test, to the local inspectors of the district where the work is being done. The local inspectors shall then satisfy themselves whether or not such process can be used with safety on the boilers of steam vessels.

In every case where repairs are to be made by these processes on the boilers of steam vessels subject to the inspection of this service, the parties making the repairs are required to notify the office of the local inspectors, in writing, giving a full detailed description of the repairs to be made, the location of the vessel, and the time the repairs are to be begun, so that inspection may be had, if practicable, prior to and during the time the work is being done.

Cracks in wrought-iron or wrought-steel headers, and cracks or sand holes in cast-steel, semi-steel, ferro-steel, malleable-iron or cast-iron headers, manifolds, crosses, tees and ells may be repaired by welding cracks or flowing metal into sand holes. Such repaired material other than headers and manifolds shall be subjected to a hydrostatic test of 3½ times the working pressure after such repairs are made. Reinforcing by building up of any of the above-mentioned articles other than headers shall not be allowed.

When crown-bar bolts have deteriorated or wasted away at top of combustion chamber under the crown bars, such deterioration not to exceed 25 per cent of the original diameter of the bolt, such bolts may be built up or reinforced by any process of autogenous welding.

Where tube sheets of boilers have deteriorated not to exceed 25 per cent of their original thickness, or where cracks have developed in tube sheets, the same may be reinforced and repaired by any process of autogenous welding, and the beading on the ends of tubes may be welded to the tube sheets by the same process. (Secs. 4403, 4418, R. S.)—"Power."

BOOK REVIEWS.

NAVAL MACHINERY.—THIRD EDITION. BY COMMANDER H. C. DINGER, U. S. NAVY. D. VAN NOSTRAND, New York, N. Y. 825 pp. and 150 illustrations, 5 folding plates.

This new (3d) edition of *Naval Machinery* is very timely in view of the great demand for marine engineering knowledge. While its name might imply that it is confined to Naval information, it is of value to all marine engineers.

The original edition of this work was compiled from articles that appeared in this Journal. The present edition has been very much enlarged and improved. It is impossible in a review to cover the scope of this book.

The material is presented in six parts: (1) Operation of Naval Machinery, which includes Getting Underway, Running Engines Underway, Accidents Underway; (2) Care and Overhaul of Main Plant, which is covered very effectually in seven chapters well divided and arranged; (3) Fittings and Auxiliaries, which is subdivided into thirteen well defined parts, each covered by a separate chapter; (4) Care and Preservation Sub-division of Hull, which covers the field outside the machinery spaces; (5) Special Auxiliary Engines, which include Steering Gear, Air Compressors, Blower Engines, Ash Hoists and Ejectors, and Workshop Machinery; and (6) Spare Parts and Tests. The last chapter on machinery tests is of special value.

This work should be on every operating engineer's book shelf. There is no better book in print for the operating engineer.

SHIP STABILITY AND TRIM. By PERCY A. HILLHOUSE, B.Sc., M.I.N.A. GIEVES PUBLISHING CO., LTD., Portsmouth, England. JOHN HOGG, London, England, 1918. 297 pp., 203 illustrations.

A discussion of displacement, stability and trim calculations without complex mathematical formulæ. Simple and accurate methods for determining these factors graphically or arithmetically are set forth.

The conditions which tend to endanger stability or affect trim in loading or operation of vessels are analysed clearly and remedial means to be observed by the master are given. The applicability of displacement metacentric and other curves furnished by the shipbuilder, and how the navigator can readily interpret such data for given conditions of loading, is explained.

The text will prove valuable to the student or the careful navigator.

CLEAR THE DECKS. By "COMMANDER." J. B. LIPPINCOTT CO. 20 illustrations, 303 pp. $1.50 net.

This is a fanciful tale of the rise of an enlisted man in the U. S. Navy to command rank. It is an intimate tale of Naval life written by one who has a keen insight and understanding of the sailor man. The tale carries one through all grades of enlisted and commissioned shipboard life, and is the "real thing."

The illustrations are actual pictures taken on board Naval vessels and depict the life of our sailors and marines thereon.